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About Authors

● **G. D. Boerlage,** *a graduate of the Technical University of Delft, Holland, first became interested in the problems of internal combustion engines in 1910-1913 when he devoted himself to the development of aviation in France and Holland. After the war he was an engineer with Messrs. Thomassen & Co., gas and Diesel engine manufacturers in Holland. Later, in 1921, he was made lector on internal combustion engines at Delft University. The Delft Laboratory (Proefstation "Delft") of the Royal Dutch Shell Group was founded under his direction in 1928. Mr. Boerlage has published a number of papers on engine research work and, together with his senior assistant, J. J. Broeze, introduced the cetene rating of Diesel fuels in 1931. He personally presented a paper at the 1932 SAE Summer Meeting.*

● **T. A. Boyd** *is head of the fuel department of General Motors Research Laboratories Section. He has joined with Campbell and Lovell in several papers dealing with the results of researches on various phases of the problem of knock in automobile engines. A number of these have been published in the SAE Journal in the past.*

● **John M. Campbell** *has been a member of the staff of General Motors Research Laboratories Section for the past ten years, concentrating his efforts on detonation, methods of eliminating it and of measuring its magnitude. In this connection he was active at the Cooperative Fuel Research Committee road tests made at Uniontown, Pa., in 1932 and 1934. He was trained in chemical engineering at the Massachusetts Institute of Technology.*

● **A. G. Cattaneo** *graduated from the Federal Technical University of Zurich, Switzerland, and later received his degree of doctor of science from the Karlsruhe University, Germany. At the latter he gained experience in research work in the Laboratory of Mechanics and Refrigeration. He joined the Delft Laboratory of the Royal Dutch Shell Group in 1934. As research engineer there he had an important part in developing the experimental quartz-window gasoline engine for the study of detonation.*

(Continued on page 42)

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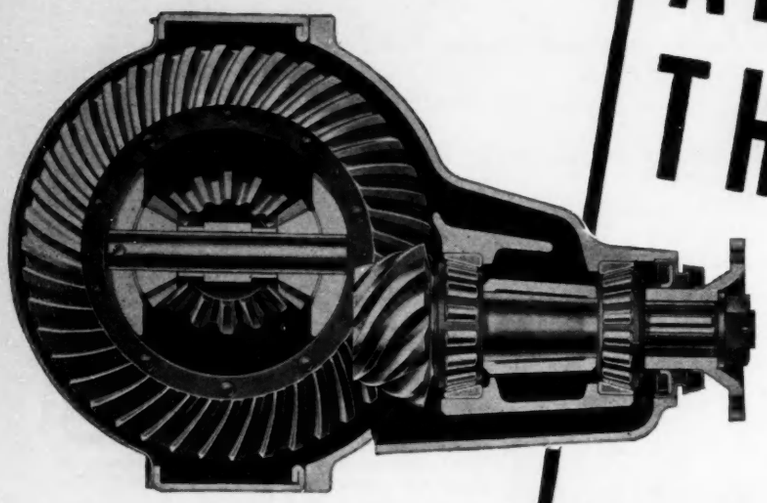

ADVANCED THINKING

Keeps TIMKEN Bearings Ahead

It has never been the policy of The Timken Roller Bearing Company to wait until the automotive industry decides upon a new development and then try to bring out a bearing application to meet it.

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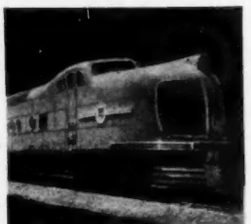

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TIMKEN

TAPERED ROLLER BEARINGS

Vehicle Design from a Maintenance and Operating Standpoint

By F. L. Faulkner
Automotive Engineer, Armour & Co.

Part 3

THE purpose of these studies by our committee is to point out inconsistencies that exist in various weight classes, with the hope that a better-balanced vehicle will be produced so that the operator in purchasing a vehicle of a certain rating can be assured of a reasonably uniform performance, irrespective of the make of vehicle.

It is obvious from a study of the 1936-37 specifications that some improvements have been made. The general rating of trucks by class is poor due to the tendency to raise the gross-vehicle-weight rating of the trucks without enlarging engines, clutches, frames, and so on.

In general there is the same wide variation in specifications within each gross class of trucks this year, as existed in 1934.

This paper is the fourth prepared by Mr. Faulkner at the request of the Transportation and Maintenance Activity Committee of the Society to present the results of studies made by the Subcommittee on Motor-Vehicle Design and Operation, of which Mr. Faulkner is chairman.

Part 1 of the study was presented at the Semi-Annual Meeting of the Society in 1934; Part 2 at the 1935 Annual Meeting. The lubrication phase of the operating problem was presented at the 1936 Annual Meeting in lieu of Part 3 presented at the 1937 Annual Meeting. Part 3 is presented as a joint effort and consists of the following five sections, of which the first three are published herewith. Sections 4 and 5 will be published later if space permits.

Section 1 – A review of the objectives and accomplishments as set forth in previous papers, with comparisons made against current production.

Section 2 – Covering the storage battery and starter-generator combinations, using data accumulated and furnished by R. M. Critchfield, chief engineer, Delco-Remy Corp.

Section 3 – "Semi-Trailer Coupling Heights," presented in its entirety as furnished by M. C. Horine, sales promotion manager, Mack-International Motor Truck Corp.

Section 4 – "Brake Application with Suggestions for Standardization," the subject matter prepared by Stephen Johnson, Jr., chief engineer, Bendix-Westinghouse Auto. Air Brake Co.

Section 5 – "Operators' Woes on Lubrication," by A. Ludlow Clayden, research engineer, Sun Oil Co.

A MEETING of the Subcommittee on Motor-Vehicle Design of the Transportation and Maintenance Activity was held June 16, 1935, at White Sulphur Springs, West Va., and as a result 18 items were presented to the Standards Committee with our recommendation for standardization, as follows:

(1) Oil-fill pipe location – left side of engine.
(2) Oil-gage location – left side of engine.
(3) Oil-fill pipe size – 2 in. minimum.
(4) Crankcase drain plug – ¾ in. by 14 pipe thread, plug recessed for ½ in. square-head wrench.
(5) Gasoline-tank location – inside cab.
(6) Gasoline-tank fill opening – external recessed and set at angle, or accessible through opening of cab door.
(7) Gasoline-tank cap – quick-action type.
(8) Gasoline-tank drain – located to drain without interference, constructed with opening flush inside tank.
(9) Radiator fill – so designed to permit filling without raising engine hood.
(10) Radiator cap – quick-action type.
(11) Radiator drain – located left side engine to permit draining without interference. Size ⅜ in., needle-seat type.
(12) Radiator mounting – flexible.
(13) Transmission-fill location – at full-lubricant level, either right or left side.
(14) Transmission-fill opening – minimum ¾-in. diameter.

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 11, 1937.]

Table 1 – Locations, Sizes, and Types of Oil, Gasoline, and Water Fills, Cab Sizes, and Accessories

OIL FILL LOCATION

LOCATION		LEFT	RIGHT	TOP	TOTAL
NO. MODELS	1934	29	0	1	30
USING	1936	22	9	1	32

RECOMMENDED STANDARD - LEFT

OIL GAGE TYPE

TYPE		BAYONET	FLOAT	OTHER TYPE	TOTAL
NO. MODELS	1934	30	0	0	30
USING	1936	32	0	0	32

RECOMMENDED STANDARD - BAYONET

OIL FILL SIZE

DIAM. INCHES		1 3/16	1¼	1½	1 5/8	2	3	TOTAL
NO. MODELS	1934	2	4	19	1	1	3	30
USING	1936	0	3	9	0	20	0	32

RECOMMENDED STANDARD -2" MINIMUM

GAS TANK LOCATION

LOCATION		UNDER CAB SEAT	FRAME REAR	FRAME SIDE	TOTAL
NO. MODELS	1934	27	1	2	30
USING	1936	28	0	4	32

RECOMMENDED STANDARD - LIGHT TRUCK - FRAME REAR
HEAVY " - UNDER DRIVERS SEAT

GASOLINE TANK FILL OPENING

LOCATION		INSIDE CAB	OUTSIDE CAB	TOTAL
NO. MODELS	1934	23	7	30
USING	1936	R.4 L.2	R.26 L.0	32

RECOMMENDED STANDARD - OUTSIDE CAB, WITH BAFFLED STANDPIPE

GASOLINE TANK FILL

DIAM. INCHES-FILL OPENING		1½	1¾	2	2 1/8	2¼	1 27/32	TOTAL
NO. MODELS	1934	4	13	1	2	8	2	30
USING	1936	2	1	28	0	1	0	32

RECOMMENDED STANDARD -2" MINIMUM

RADIATOR FILL

TYPE FILL CAP		THREADED	BAYONET	OTHER TYPE	TOTAL
NO. MODELS	1934	6	23	1	30
USING	1936	0	32	0	32

RECOMMENDED STANDARD-BAYONET TYPE, EXTERNAL OF HOOD

RADIATOR FILL

DIAM. FILL OPENING "		1½	1¾	1 17/32	1¾	1 7/8	2	2¼	2½	TOTAL
NO. MODELS	1934	2	3	3	4	1	14	1	2	30
USING	1936	0	0	0	0	0	2	1	29	32

RECOMMENDED STANDARD - 2½"

RADIATOR MOUNTING

TYPE		RUBBER	SPRING	OTHER	TOTAL
NO. MODELS	1934	6	20	4	30
USING	1936	32	0	0	32

RECOMMENDED STANDARD - RUBBER

LUBRICATION FITTINGS

TYPE		ALEMITE	ZERK	"SNAP-ON" ZERK	TOTAL
NO. MODELS	1934	4	17	19	41
USING	1936	0	0	32	32

RECOMMENDED STANDARD -"SNAP-ON" ZERK

TWO MAN CAB

DIMENSIONS INCHES	VARIATION IN 30 MODELS 1934	VARIATION IN 32 MODELS 1936	RECOMMENDED STANDARD
INSIDE HEIGHT	46 to 57	45 to 57¾	50
INSIDE WIDTH	49¼ to 68½	46½ to 57¾	50
INSIDE BACK TO DASH	28 3/8 to 55	30 to 37	36
FLOOR TO TOP SEAT CUSHION	12½ to 17	10 to 15	13
FLOOR TO STEERING WHEEL	21½ to 26	19 to 25¼	22
DOOR WIDTH	26 to 33½	27 3/8 to 33	31
DOOR HINGED, FRONT REAR		ALL FRONT	FRONT

WINDSHIELD WIPERS

TYPE		VACUUM	ELECTRIC	MECH. POWER	TOTAL #
NO. MODELS	1934	26	4	0	30
USING	1936	32	0	0	32

RECOMMENDED STANDARD - 2 WIPERS, VACUUM OR ELECTRIC

(15) Transmission-drain opening – minimum ¾-in. diameter.
(16) Transmission drain plug – size, ¾-in. diameter. Countersunk type for ½-in. square-head-type wrench.
(17) Rear-axle fill opening – ¾-in. diameter, countersunk for ½-in. square-head-type wrench.
(18) Rear-axle drain plug – ¾ in. by 14 pipe thread, countersunk for ½ in. square-head wrench.

A comparison of locations of oil, gas, and water fills, cab sizes, accessories, and so on, on 30 models of 1934 trucks as compared with 32 models of late 1936 and early 1937 trucks shows, in the 32 current models checked, that oil fills and gages on the left side of the engine and gas fills on the right side of the cab is the prevalent trend, as compared to 1934 models which, in general, had both fills on the left side. There is no definite improvement in the size of the oil and gas fill-pipe openings as recommended by the operators in 1934. These data are given in Table 1.

There has been little change in the type or location of radiator fill openings from the 1934 models, except that a few manufacturers are now putting the opening under the hood, contrary to the recommended 1934 standard. However, all the new models that were checked have rubber-mounted radiators, which follows the recommended standard and is a distinct change in the right direction from previous models. Likewise, the standardization of all models on Alemite "snap-on" Zerk lubrication fittings is in accord with the recommended standard of 1934. Two-man cabs have approximately the same dimensions as the 1934 models. In many cases the dimensions are smaller than the recommended standards. One vacuum-type windshield wiper is still the general standard, with the second wiper an optional extra.

Late in 1935 in a paper, "Trends in Automotive Servicing," by Coombs and Veeder, Shell Petroleum Corp., certain recommendations regarding standardization of service points on cars were listed. These recommendations, I am advised, have been approved by the American Petroleum Institute Committee on marketing equipment.

Their recommendations follow very closely the recommendations of our committee. It is gratifying to me to see the service people, other than operators, interesting themselves in this problem.

The question logically asked is – what is the manufacturer of motor vehicles doing about it? Then, turning to a survey of the product for the answer, we find that a serious effort is

being made on the part of some of the manufacturers to clean up their designs, particularly from the accessibility standpoint. Whereas, on the other hand, we find too many examples of poor design from the maintenance standpoint.

What does the operator think of the job the manufacturer is doing? I quote: "Inspection of the new 1937 passenger models indicates clearly that the trend in design is without regard for accessibility." Another operator says: "An examination of the engine shows the cooling-system drain cocks to be barely accessible when the engine is out on a motor stand, and evidently they could not be reached in the chassis without getting under the car." Another states: "To remove the starting motor on 1935 and 1936 models, it is necessary first to remove the distributor and disconnect oil-filter lines. After replacing the starter, it is then necessary to re-time the distributor and reconnect the oil lines."

A certain bus operator states: "With the rear-engine type that we are now purchasing we find one make that requires the removal of the engine, clutch, and transmission of the unit whenever it is necessary to make repairs to either clutch or transmission."

A large operator of light-duty vehicles states: "The distributor is in close quarters and has to be removed to clean or change points. Valves are difficult to remove. Either motor or rear end has to be removed to change the clutch or transmission. Also spring-hanger bushings are difficult to remove and install."

A highway-transport operator states: "The location of batteries covered with floor boards and floor mats is hard to understand. Manufacturers incorporate time-savers in their factories, but they don't think much about the customers, whether fleet operator or motorist, because they continue to hide the battery in the most inaccessible places, then recommend sending them in for monthly inspection. By incorrect installation they make a 5-min. job into a 25-min. operation, sometimes longer."

I might add at this point that it is gratifying to see one of the major manufacturers now installing the battery of both cars and trucks in a very accessible place.

Another operator states on the subject of accessibility: "Sometimes I think that the designs with respect to accessibility are getting worse. Take for instance brake shoes, they have been housed so completely as to make the inspection of brake lining and brake-shoe clearances an impossibility without first pulling wheels, and I don't care how simple that operation may be considered to be, nevertheless, it is costly."

Another operator complains about radiator and other cooling-system connections: "Hose clamps we call them, why don't we get some improvement there? We experience water leaks just because some five-cent clamp connection has gone haywire. With radiators now so little related to the shell, the radiator should no longer have to be shifted for hood fittings, consequently standard-length union-type connections might well be used."

Another bus operator complains about inaccessibility of the oil-filler neck on the accessory housing on a certain model bus: "One would think that an item as important as lubrication of air compressor, water pump, and generator, oil pump and fan, all carried in this accessory housing and requiring checking daily, could be made as accessible as a radiator filler neck. However, it is necessary to lie on one's back and use the hooked-nose hand pump to fill this housing."

Other operators are complaining about the absence of drain plugs in the rear-axle housing on so many 1937 cars, also of clutch and brake-pedal lubrication, with fittings poorly located or inadequate. Certain makes of shock absorbers require complete removal for refilling. This operation is recommended each 5000 to 10,000 miles. Certain universal joints are designed without any provision for lubrication without dismantling the joints. Certain of the fan installations are more difficult to lubricate and adjust than others. Valve tappets and other accessories are difficult to reach on many makes. And last, but not least, is the matter of lubrication on hypoid axles wherein, if the manufacturers' recommendations are followed, it will require a highly questionable type of compound, called hypoid lubricant, to be kept in all garages to service these special axles, and to be used for no other purpose, as warned by the car manufacturers.

Our committee recommended, among other things, that all gasoline-fill openings, whether car or truck, should be on the left-hand side of the vehicle. The reasoning behind this recommendation is obvious. I had occasion to be at the factory of a certain manufacturer recently and, in looking over the 1937 models, noticed that the gasoline-tank filler was on the right side of the car whereas on last year's models, it was on the left. Upon asking why this change had been made, no straightforward explanation could be had, the closest approach being, however, that in the present model, due to the concealed spare tire, there was some interference and, rather than change the tire mounting slightly, it was simpler to move the gasoline tank opening from the left to the right side of the car. I cite this case as an example of how little thinking really is applied to the servicing problem by the manufacturer.

For the benefit of those who are not familiar with my previous papers on the subject of motor-vehicle design, permit me to review briefly that phase of the subject pertaining to truck ratings from the standpoint of a comparison of the strength of their component parts.

It was brought out that motor-truck ratings had been discussed pro and con over a period of years without anything tangible being developed that the operator could use as a guide in the purchase of motor vehicles for a definite transportation job. I attempted to correlate certain data furnished by the manufacturers by using their own advertised ratings to show first, the lack of consistency, and second, the lack of standardization in the industry.

The method used in developing this practical rating was simply to take the manufacturer's advertised gross vehicle weight with his advertised standard axle ratio, equipping that vehicle with a sufficient tire size to comply with the Rubber Association's rating for the particular gross vehicle weight, and for the performance factor compute the grade ability at 20 m.p.h.

As a check on the strength of the component units in the vehicle, the front-axle knuckle diameter at the inside bearing was taken; the rear-axle outside-tube diameter, full-floating type; section modulus of frame; clutch area; effective braking area; and engine displacement, all computed in terms of 1000 lb. of advertised gross vehicle weight, maximums, minimums, and averages for all weight classes of vehicles.

These data were presented graphically and many inconsistencies were shown.

In order that you may have before you the 1934 data, I have plotted the 1934 averages against the maximums, minimums, and averages of the 1936 production in Figs. 1 to 9. The data are tabulated in Table 2.

Referring to Fig. 1 showing chassis weight-price comparisons, all price information used is taken from the August, 1936, issue of the *Commercial Car Journal,* list price less national fleet discount. All dimensions, weights, and gross

Table 2 – Data Comparing Price-Weight Ratios, Specifications, and Grade Abilities

8000# to 10000# ADVERTISED G.V.W.

Make and Model	*Chassis Price Per 1000# Chassis Weight $	RATIO OF SPECIFICATIONS TO G.V.W. ÷ 1000							% Grade At 20 M. P. H.	
		*Chassis Price $	Piston Disp. Cu. In.	Clutch Area Sq. In.	Brake Area Sq. In.	Frame Section Modulus	Diam. Front Axle Knuckle	Diam. F.F. Rear Axle Tube In.	Calculated	Reported
1	175	48	21.2	9.92	19.95	.437	.151	.368	3.55	3.55
2	179	53	22.0	9.93	22.35	.522	.159	.303	4.10	4.10
3	188	72	23.4	14.55	29.40	.705	.180	.309	4.20	4.40
Max.	188	72	23.4	14.55	29.40	.705	.180	.368	4.20	4.40
Min.	175	48	21.2	9.92	19.95	.437	.151	.303	3.55	3.55
Avg.	181	58	22.2	11.47	27.23	.555	.163	.327	3.95	4.02
1934 Avg.	173	64	24.56	11.41	27.31	.577	.159	.348	4.44	

10000# to 12000# ADVERTISED G.V.W.

Make and Model	*Chassis Price per 1000# Chassis Weight $	RATIO OF SPECIFICATIONS TO G.V.W. ÷ 1000							% Grade at 20 M. P. H.	
		*Chassis Price $	Piston Disp. Cu. In.	Clutch Area Sq. In.	Brake Area Sq. In.	Frame Section Modulus	Diam. Front Axle Knuckle	Diam. F.F. Rear Axle Tube In.	Calculated	Reported
4	170	46	19.82	9.09	14.32	.423	.286	.363	3.52	4.00
5	193	54	19.80	8.18	21.40	.430	.130	.260	3.30	3.30
6	156	44	18.90	8.18	22.80	.422	.125	.347	3.10	3.10
7	149	43	19.38	8.86	30.60	.441	.127	.227	3.43	3.50
8	171	48	18.50	8.20	25.30	.446	.131	.348	3.86	4.02
9	150	40	18.30	8.50	29.20	.432	.125	.301	3.38	4.85
36	203	77	19.90	8.95	24.25	.653	.300	.251	4.75	5.00
Max.	203	77	19.90	9.09	30.60	.653	.300	.363	4.75	5.00
Min.	149	40	18.30	8.18	14.32	.422	.125	.227	3.10	3.10
Avg.	170	50	19.23	8.52	21.98	.464	.175	.298	3.62	3.97
1934 Avg.	191	63	21.24	9.92	21.88	.552	.142	.316	3.68	

12000# TO 14000# ADVERTISED G.V.W.

Make and Model	*Chassis Price Per 1000# Chassis Weight $	RATIO OF SPECIFICATIONS TO G.V.W. ÷ 1000							% Grade at 20 M. P. H.	
		*Chassis Price $	Piston Disp. Cu. In.	Clutch Area Sq. In.	Brake Area Sq. In.	Frame Section Modulus	Diam. Front Axle Knuckle	Diam. F.F. Rear Axle Tube In.	Calculated	Reported
10	216	60	17.90	9.73	23.80	.643	.120	.296	3.31	3.31
11	222	62	19.80	9.73	21.40	.443	.111	.213	3.05	3.16
12	210	53	19.50	9.14	15.00	.402	.254	.316	3.82	4.12
13	172	48	17.78	7.85	28.10	.463	.117	.219	3.00	3.00
14	198	56	18.40	9.55	23.90	.456	.120	.202	2.90	2.90
15	199	57	17.80	8.92	23.25	.347	.130	.340	4.20	4.34
16	140	36	17.70	9.90	28.00	.432	.240	.300	1.90	3.33
17	190	52	21.90	10.30	20.60	.500	.130	.118	3.10	3.57
Max.	222	63	21.90	10.30	28.10	.643	.254	.340	4.20	4.34
Min.	140	36	17.70	7.85	15.00	.347	.111	.202	1.96	2.90
Avg.	193	54	18.85	9.39	23.01	.461	.153	.263	3.20	3.47
1934 Avg.	217	72	19.53	9.74	23.16	.531	.127	.246	3.37	

14000# to 16000# ADVERTISED G.V.W.

Make and Model	*Chassis Price Per 1000# Chassis Weight $	RATIO OF SPECIFICATIONS TO G.V.W. ÷ 1000							% Grade at 20 M. P. H.	
		*Chassis Price $	Piston Displ. Cu. In.	Clutch Area Sq. In.	Brake Area Sq. In.	Frame Section Modulus	Diam. Front Axle Knuckle	Diam. F.F. Rear Axle Tube In.	Calculated	Reported
18	250	72	17.85	8.75	23.30	.446	.153	.192	3.15	3.06
19	203	58	17.10	8.86	24.40	.424	.112	.188	2.38	2.38
20	300	107	17.18	9.25	22.80	.520	.117	.175	2.56	2.68
21	223	70	15.96	7.96	24.20	.525	.116	.304	3.43	3.47
22	201	58	16.80	8.64	17.80	.690	.111	.249	3.30	3.48
23	256	87	19.25	8.64	22.00	.388	.245	.178	3.51	3.58
37	256	83	17.45	9.35	21.30	.490	.225	.187	3.60	3.60
Max.	300	107	19.25	9.35	24.40	.690	.245	.304	3.60	3.60
Min.	201	58	15.95	7.96	17.80	.388	.111	.175	2.38	2.38
Avg.	241	76	17.65	8.63	22.26	.496	.151	.210	3.13	3.18
1934 Avg.	318	115	16.60	11.07	25.79	.454	.112	.207	3.08	

16000# to 18000# ADVERTISED G.V.W.

Make and Model	*Chassis Price Per 1000# Chassis Weight $	RATIO OF SPECIFICATIONS TO G.V.W. ÷1000							% Grade At 20 M.P.H.	
		*Chassis Price $	Piston Displ. Cu.In.	Clutch Area Sq.In.	Brake Area Sq. In.	Frame Section Modulus	Diam. Front Axle Knuckle In.	Diam. F.F. Rear Axle Tube In.	Calculated	Reported
24	262	83	17.58	11.11	26.25	.460	.213	.300	3.25	3.80
25	273	78	14.60	7.92	22.02	.446	.100	.187	2.38	2.38
26	274	95	17.20	9.15	23.30	.573	.109	.256	3.02	3.14
27	314	109	16.88	7.73	23.00	.412	.213	.187	3.00	3.21
Max.	314	109	17.58	11.11	26.25	.573	.213	.300	3.25	3.80
Min.	262	78	14.60	7.73	22.02	.412	.100	.187	2.38	2.38
Avg.	281	91	16.59	8.98	23.64	.473	.159	.225	2.91	3.13
1934 Avg.	303	105	18.36	9.86	24.04	.527	.107	.181	3.39	

18000# to 20000# ADVERTISED G.V.W.

Make and Model	*Chassis Price Per 1000# Chassis Weight $	RATIO OF SPECIFICATIONS TO G.V.W. ÷10000							% Grade At 20 M.P.H	
		*Chassis Price $	Piston Displ. Cu. In.	Clutch Area Sq. In.	Brake Area Sq. In.	Frame Section Modulus	Diam. Front Axle Knuckle In.	Diam.F.F. Rear Axle Tube In.	Calculated	Reported
28	282	92	19.65	9.76	25.05	.404	.189	.288	3.90	4.40
29	250	67	17.15	15.72	22.18	.477	.111	.187	2.56	2.56
30	288	85	15.45	12.56	19.95	.480	.167	.100	3.17	3.55
31	270	76	15.48	8.69	21.08	.484	.153	.094	2.65	2.65
32	300	61	14.52	6.94	19.45	.500	.100	.158	2.20	2.20
33	339	118	17.20	7.72	20.58	.300	.069	.180	3.26	3.50
34	415	155	15.48	8.07	21.55	.295	.072	.158	2.40	2.50
35	346	125	16.65	8.29	25.90	.422	-	.187	2.45	2.50
38	336	116	17.20	9.62	17.75	.422	.198	.142	2.67	3.00
39	372	125	18.50	9.12	16.90	.400	.185	.167	3.10	3.10
Max	415	155	19.65	15.72	25.90	.500	.198	.288	3.90	4.40
Min	250	76	14.52	6.94	16.90	.295	.069	.094	2.20	2.20
Avg	320	104	16.75	9.45	20.60	.413	.122	.160	2.81	2.94
Avg.1934	364	119	17.31	9.61	20.92	.573	.097	.185	3.55	

* Chassis Prices Based on "Commercial Car Journal" August, 1936, Less National Fleet Discount.

ratings used were furnished by the respective manufacturers.

A study of the weight-price comparisons indicates definitely that the average manufacturer is giving more per dollar than in 1934 in the face of the continued rise of both material and labor costs and, on behalf of the operators, I wish to take this opportunity to congratulate the manufacturers on this remarkable accomplishment.

Fig. 2 speaks for itself.

There has been considerable discussion regarding engine sizes for various weight-class vehicles, and it appeared from the 1934 data that there was a trend toward larger engines to meet the definite grade-ability requirements in certain States. However, it will be noted in referring to Fig. 3 that the average engine size for every weight class is materially less than found in the same weight classes of 1934.

The foregoing reduction is brought about undoubtedly not by reducing engine size, but on account of increasing gross vehicle weight without increasing the engine size correspondingly.

The clutch problem seemingly never will be solved. Operators as a class have complained about clutch design and clutch troubles as far back as I can recall. Rarely do a group of operators discuss their maintenance problems that clutches do not come in for their share of the discussion. It was brought out two years ago that there was a material deficiency in clutch sizes, and it was hoped that after that discussion, larger and more efficient clutches would be forthcoming. However, upon examination of the data as shown in Fig. 4, we find that clutches are now called upon to do more work per square inch of area than was imposed upon them in 1934. In discussing this clutch problem with a reputable clutch manufacturer, he stated that it was a very simple matter for him to furnish clutches that would stand up, but

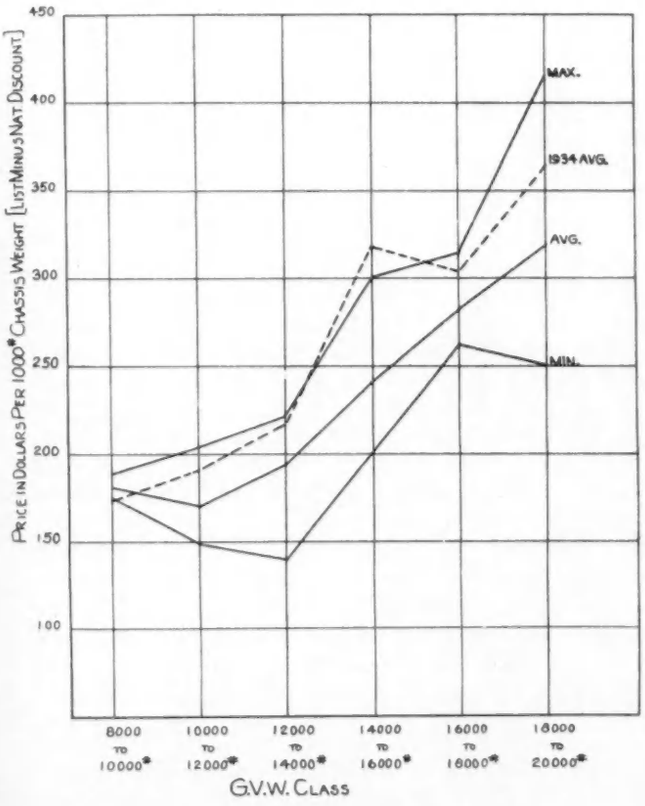


Fig. 1 – Chassis Weight-Price Comparisons

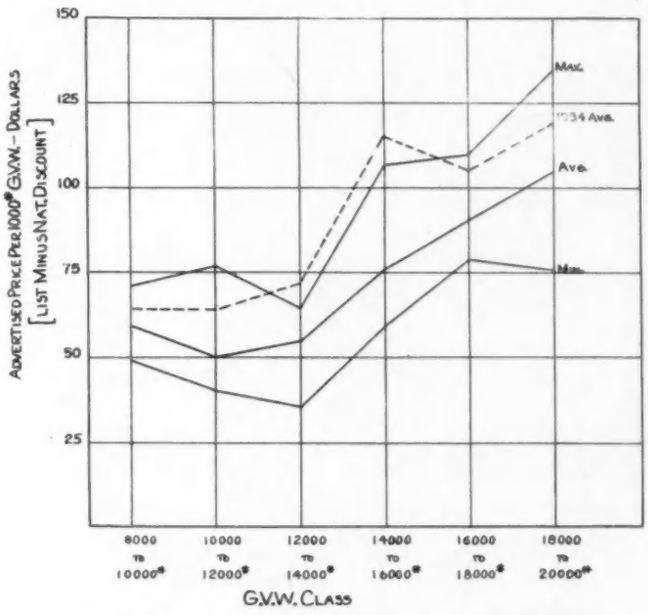


Fig. 2 – Gross Vehicle Weight-Price Comparisons

it was always a matter of price. Just why a manufacturer is willing to cut to the bone a part that is as overworked as the clutch and so difficult to repair when it fails, is hard for the operator to understand. There is a material item of only a few cents for facing, but a labor charge of several dollars.

Improvement is noted in the braking systems, not only from the standpoint of increased effective braking area, but also in the actual performance of the brakes themselves. In the 8 to 10,000; 10 to 12,000; and 12 to 14,000 lb. gross-vehicle-weight classes, with increases in the rating, brake area has kept step, as shown in Fig. 5. Apparently the manufacturers have learned that 38.6 per cent of their vehicles were rejected during three months of last year's compulsory vehicle inspection.

It is assumed generally that the section modulus of a frame is indicative of its strength, all other factors being equal. In examining the data as shown in Fig. 6, it is worthy of note that improvement is shown in only one weight class, namely, 14 to 16,000 lb. gross vehicle weight. It is of particular interest to note that considerable improvement is made in the 14 to 16,000 lb. class all along the line, and if you will recall, this is the particular weight class that was pointed out in the 1934 data as being virtually a clean-up model for the 12 and 16,000 lb. classes.

In examining the front-axle data we note in practically every model studied that marked improvement has been made not only in front-axle location, better design, but also in increased strength in the load-carrying members as well. With higher operating speed and greater rate of weight transfer, the front axle was rapidly becoming a troublesome member in the chassis. Hard steering soon developed due to poor alignment and to deformation of parts under high stress. It will be noted in referring to Fig. 7 that the minimum diameter of the front-axle knuckle found today is practically the average of 1934, whereas the average in 1936 production is materially higher with outstanding maximums in many offerings. What we note in front-axle design was hoped for in clutches and anticipated in braking systems.

Little change is noted in rear-axle capacities as indicated by data shown in Fig. 8. Using the same axle in a given

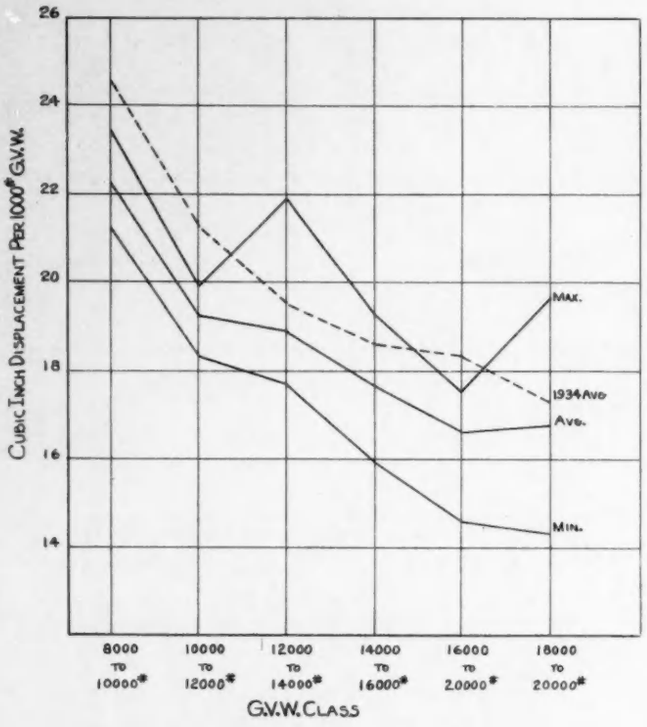


Fig. 3 – Piston-Displacement Comparisons

model having a higher gross vehicle weight than previously rated, obviously reflects adversely in this comparison. However, it must be kept in mind that changes in load distribution have shifted some of the burden from the rear axle to the front, all of which may in part compensate for the apparent deficiencies noted.

In compiling the grade-ability data as shown in Fig. 9, many discrepancies were found in the reported data furnished by the manufacturers on account of the fact that a great deal of the information was based on the most favorable axle ratio. I have shown one average curve constructed from data as reported. The maximum, minimum, and average curves shown were computed on the basis of standard axle ratios with the proper tire size to carry the rated gross vehicle weight (G.V.W.) according to the Rubber Association's ratings.

You will note, as to be expected, with increased load rating without increasing engine size, that the average grade ability on the 1936 models is lower with one exception than reported from 1934 models. This exception is in the much-discussed 14 to 16,000 lb. class which, we have shown all along the line, is a materially improved vehicle.

In summarizing the preceding data, it was obvious from a study of the specifications that, although there has been some improvement in the size of some truck parts, such as front axles, brake drums, and so on, the general rating of trucks by class is poor, due to the tendency to raise the gross-vehicle-weight ratings of the unit without enlarging engines, clutches, and so on, in proportion. The specifications that show the greatest decrease per 1000-lb. G.V.W. rating, are engine size, clutch area, frame strength, and brake area. This reduction is not caused by a decrease in the strength or size of the specifications as used in 1934, but by the abnormal increase in the manufacturer's G.V.W. rating. In general there is the same wide variation in specifications within each gross class of trucks this year as existed in 1934. The purpose of these studies by our committee is to point out inconsistencies that exist in various weight classes, with the hope that a better-balanced vehicle will be produced so that the operator, in purchasing a vehicle of a certain rating, can be assured of a reasonably uniform performance irrespective of the make of the vehicle.

When the 1934 data were presented, considerable comment was expressed by various manufacturers' representatives that this type of study had a tendency to standardize truck design, and that you cannot standardize on one truck design to fit all needs. Also, that truck manufacturers should design their vehicles to meet individual requirements. I was particularly interested to note the comments on this discussion in the January, 1935, issue of *Automotive Industries,* from which I quote: "Somehow this discussion seems reminiscent of discussions on this subject a number of years ago except that the situation was reversed at that time, with operators demanding special designs to fit their needs and manufacturers holding out for standardization – the exact reverse of last week's discussion. Mr. Faulkner further emphasized that the objective of the study was largely to provide a measuring stick on which the comparative values of trucks on the market can be judged, and that for this purpose a comparative study of existing trucks provided the best possible means of solution."

The operator's position today is the same as at that time. We are not desirous of designing or building trucks but, when we are confronted daily with manufacturers' errors that we must live with day in and day out, we feel that we are entitled to a hearing on the subject. Until such a time as

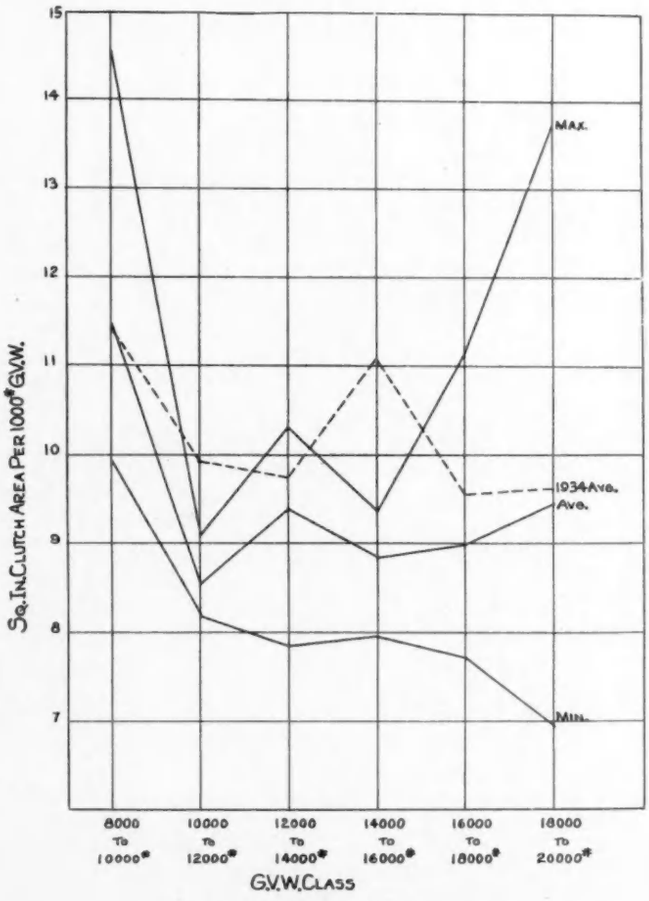


Fig. 4 – Clutch-Area Comparisons

a better method of evaluating a motor truck is produced, I will continue to use and propose the use of the foregoing method.

Section 2

Battery and Starter-Generator Combinations

Much has been said regarding the deficiencies existing in the electrical equipment on the modern motor vehicle, including batteries, starters, generators, and ignition sets. I have spent considerable time in attempting to analyze the deficiencies in the ignition equipment, attempting to establish a point from which we could discuss intelligently what is needed for a certain type of duty. The difficulties encountered in 1934 are prevalent today and, in many instances, they are aggravated. It was pointed out that, in the high-speed, high-compression engines where the ignition job is the most difficult, we still find the poorest type of equipment. And

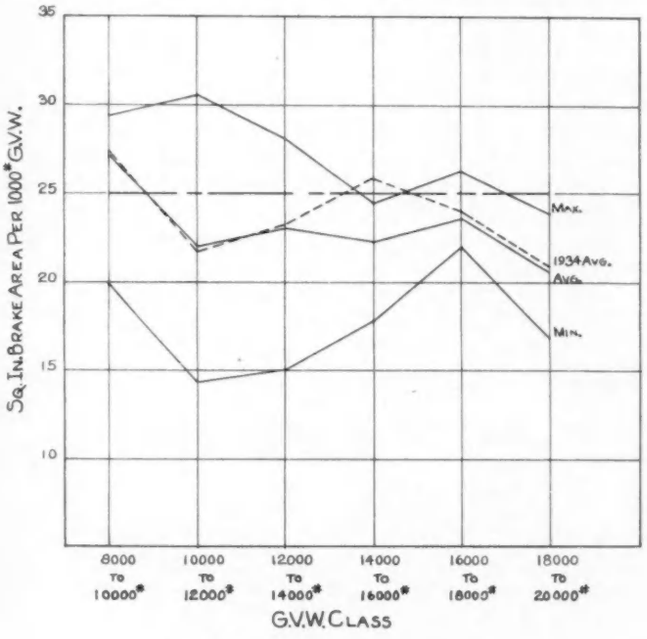


Fig. 5 – Brake-Area Comparisons

also, when it is recognized that the majority of our fleets today are made up of a large percentage of light-duty equipment, the maintenance problem with which we are confronted can be appreciated. In many instances repair expense accumulated throughout the life of the vehicle on the ignition equipment alone, including frequent tune-ups, overhauling, and replacement of worn parts, equals the repair expense on the remainder of the engine. In general, coils, condensers, and distributors furnished as standard equipment on the light-duty vehicles are not satisfactory for the job. I have discussed this problem many times with equipment manufacturers, and there is no lack of knowledge on their part as to what is necessary to eliminate the difficulty. However, again they are confronted with the problem of building equipment to a price level that will just permit getting by for a limited period of time.

In view of the fact that engine failures, which are expensive and conducive to higher operating costs, due to ignition faults are the most common cause of delay, would it not appear logical that more thought should be given to, and less compromises made with, the ignition equipment?

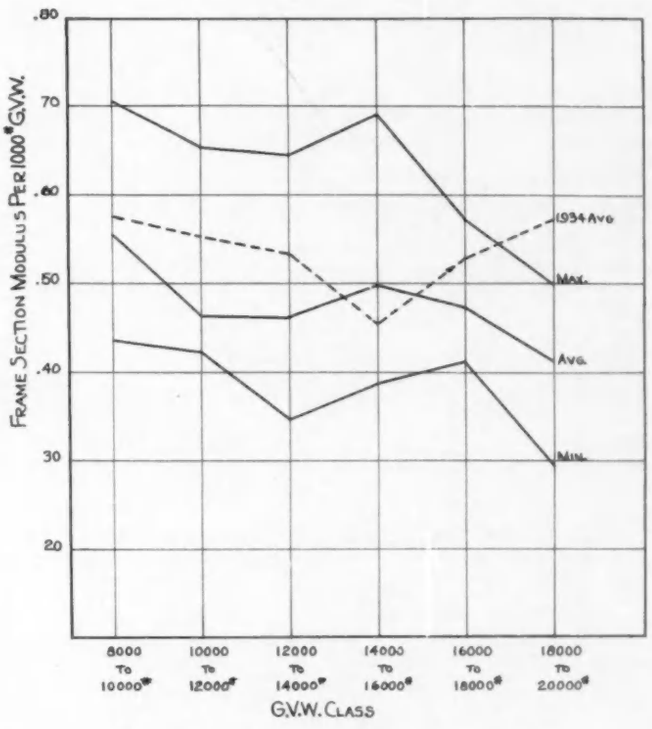


Fig. 6 – Frame Section-Modulus Comparisons

I do not see any possibility at this time of making any recommendations towards standardization of ignition equipment.

The Storage Battery

I have mentioned earlier in this paper the utter disregard paid by the average manufacturer to the location and mounting of the storage battery. Probably the old adage: "out of sight out of mind," applies, as apparently the battery is not being given the attention it warrants.

In checking over the battery specifications of some 38 truck

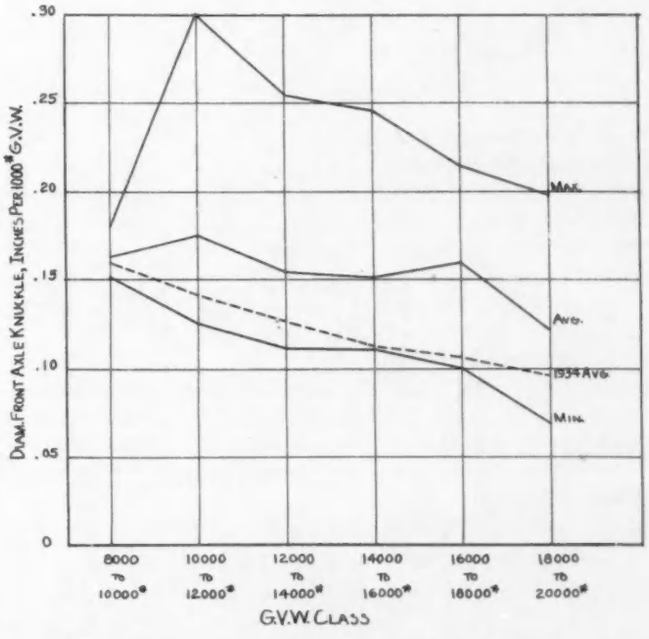


Fig. 7 – Diameter of Front-Axle Knuckle Comparisons

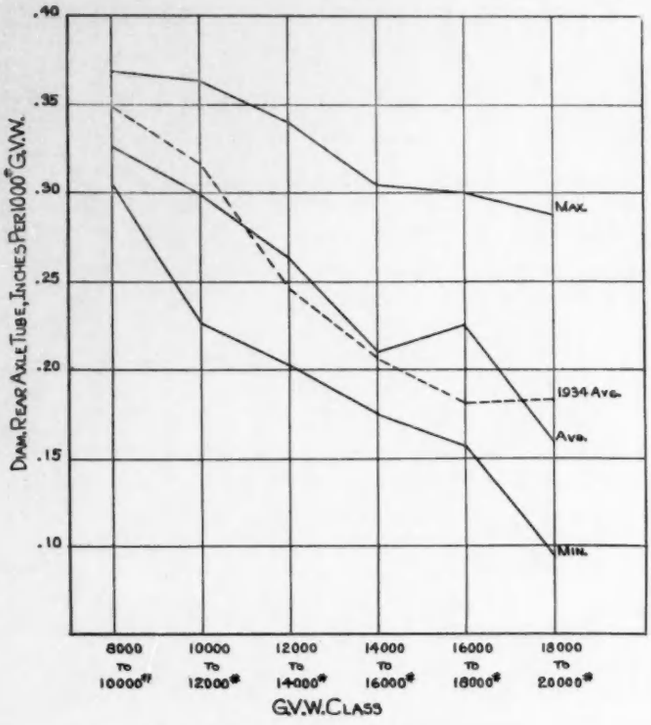


Fig. 8 – Rear-Axle Tube Comparisons

models this year, we were startled to find that 29 of these batteries are rated under the old S.A.E. rating, 8 under the new rating, and 1 not rated or shown on the chart, it having special characteristics.

In order to give you this battery story, it is necessary that we use both S.A.E. ratings in connection with the charts.

Referring to Fig. 10, showing S.A.E. rated batteries used for various cubic-inch displacement engines, you will note a marked improvement in battery size and rating as compared to the data shown in the 1934 comparison. Out of 39 models studied at that time using engines up to 360 cu. in. displacement, there were only 8 that furnished a battery as standard equipment that had an S.A.E. rating of over 3 min., at the 300-amp., 0 deg. fahr. rating. In the 1936 data we only find one manufacturer that supplies a battery lower than the 3-min. rate. Although the improvement shown is encouraging, there is still room for material improvement.

From field experience we feel that a 3-min. rated battery on engines from 200 to 250 cu. in. displacement is the minimum battery size that we can get along with satisfactorily. It would, therefore, appear logical to assume that, if the engine size is increased, we could anticipate proportionately larger capacity batteries being furnished, and I have shown in Fig. 10 a theoretical curve constructed on the basis of an ideal battery capacity for any given engine size.

Starting Motors

Seemingly little improvement has been made in starting-motor characteristics from the standpoint of eliminating starting difficulties under low-temperature operating conditions. I pointed out in the 1934 data that it was quite obvious that the manufacturers do not know the effort required to crank the engine at starting speed at 0 deg. fahr. Referring to Table 3, you will note that the manufacturer has been unable to furnish this information, except in a very few cases. Likewise, if you examine the data you will find many inconsistencies. For example, starter having a torque of 14 ft-lb., with an armature ratio to flywheel of 11.2, produces a maximum torque at flywheel of 156.8 ft-lb.; yet the manufacturer states that this engine requires 175 ft-lb. starting

Table 3 – Starting Characteristics

#	Displ.	Main and Rod Bearing Area Sq. In.	Comp. Ratio	Starting Motor Torque Ft. Lb.	Minimum Cranking Speed to start R.P.M.	Ratio Armature to Flywheel	# Torque at Flywheel Ft. Lb.	Starting Torque Required Ft. Lb. 0°F.	Starting Torque Required Ft. Lb. 32°F.	S.A.E. Oil Req. 0°F.	S.A.E. Battery #	SAE Min. at 300 Rtg.	Minimum Temp. Battery Will Start Eng. °F.
a	201.3	73.06	5.8	12	-	16.22	194.6	-	-	20W	1 *	3.1	-
b	209	57.21	5.4	12.5	150	11.3	141	-	-	10W	1 *	3.0	-30
c	209.5	28.6 P	5.75	19	-	11.2	213	-	-	30	2H	4.2	-
d	213.2	30.4 P	5.7	12	20-30	10.5	126	107	-	10	1 *	2.1	-15
e	213.3	29.7	6.0	12	-	16.1	193	-	-	10W	1 *	3.0	-
f	217.7	79.5	5.6	12	-	16.22	194.6	-	-	20W	1 *	3.1	-
g	217.8	31.56	6.0	15	35	14.4	216.0	-	-	10	1 *	4.1	-10
h	221	62.5	5.3	14	40	11.2	156.8	175	160	20	-	-	-20
i	222.7	31.3	5.4	12	20-30	10.1	121.2	103	-	10	1 *	3.0	-15
j	228	107.8	5.4	12.5	150	16.7	208.7	-	-	10W	1 *	3.0	-30
k	238.9	29.7	5.1	12	-	13.67	164	-	-	10W	1 *	3.0	-
l	241.5	94.2	5.4	12	-	16.22	194.6	-	-	20W	2 *	4.3	-
m	244	32.6 P	5.4	19	-	11.2	213	-	-	30	2H	4.2	-
n	257.5	37.5	4.9	16	-	13.67	219	-	-	10W	1 *	3.3	-
o	262.9	44.9	5.3	16	35	11.2	179	-	-	20	4 *	5.4	-10
p	268	107.8	5.58	15	150	16.7	250	-	-	10W	1 *	3.0	-30
q	270	38.4	5.5	15	-	12.5	187	-	-	30	1 *	3.1	-20
r	278.7	51.1	4.66	22	20-30	11.5	253	129	-	20	2A	4.05	-15
s	281.7	44.9	5.25	16	-	12.7	204	-	-	20	4 *	5.4	-
t	286	37.5	4.9	16	-	13.67	219	-	-	10W	1 *	3.3	-
u	298	51.2	5.7	22	20-30	11.5	253	129	-	10	2A	4.05	-15
v	303	48.3	5.5	15	-	12.5	187	-	-	30	1 *	3.1	-20
w	309	107.8	5.44	16.5	150	13.6	224	-	-	10W	2 *	4.3	-30
x	309.6	148.6	5.4	24	-	13.77	330	-	-	20W	3 *	5.4	-
y	318	38.3	5.5	15	-	12.5	187	-	-	30	1 *	3.1	-
z	318.4	59.8 P	5.48	22.5	-	12.6	284	-	-	30	3H	5.3	-
A	331	47.07	4.74	16	-	13.62	218	-	-	10W	1 *	3.3	-
B	358.1	51.50	5.1	22	-	11.54	253.8	-	-	30	5 *	6.6	-
C	360.6	59.87 P	5.4	22.5	-	12.6	284	-	-	30	3H	5.3	-
D	380.9	64.4 P	4.75	26	-	12.54	326	-	-	30	4H	6.4	-
E	396	61	5.0	19	-	10.62	202	-	-	30	4 *	5.4	-20
F	400.9	43.58	4.75	16	-	13.67	218	-	-	10W	1 *	3.3	-
G	404	68.86	5.5	22	30	18.15	399.3	340	142	40	4 *	6.1	-
H	427.5	64.4	4.6	26	-	13.80	358	-	-	30	4H	6.4	-
I	428	137.2	4.75	19	150	9.7	184	-	-	10W	2 *	4.3	-30
J	450.9	43.58	4.73	16	-	13.67	218	-	-	10W	1 *	3.3	-
K	463	68.8	5.5	22	35	18.15	399.3	360	150	40	4 *	6.1	-
L	460	61.0	5.0	19	-	10.62	202	-	-	30	3 *	5.1	-

*Old S.A.E. Ratings P – Projected Area
#Torque at Flywheel – Starter Stall Torque x Armature-to-Flywheel Gear Ratio

torque at 0 deg. fahr., and 160 ft-lb. of torque at 32 deg. fahr. From the data given it would appear that this engine could not be started at 32 deg. fahr. Although they state further that the battery furnished with this starter combination will start the engine at —20 deg. fahr.

Less than half of the manufacturers reporting know the minimum cranking speed to start the engine. It is interesting to note that engines *b* and *j* in Table 2 require a cranking speed of 150 r.p.m. to start; yet you will note that the minimum temperature that the battery will start these engines is at —30 deg. fahr., using a No. 1 old S.A.E. rated battery, the lowest in the S.A.E. group. The torque at the flywheel, expressed in foot-pounds, was computed for this table from the starter stall torque, multiplied by the armature-to-flywheel gear ratio.

Generators

Our study of generator characteristics shown in Table 4 reveals that there has been a marked improvement in generator design since our previous review. With the exception of generators *e*, *n*, and *t*, we find a very satisfactory maximum current output under high-temperature operating conditions. This improvement has been brought about by providing better cooling for the generator, a better balance between the copper-to-iron ratio, and a more liberal use of heat shields.

Let us refer to the *J* combination, where an engine of 450.9 cu. in. displacement, requiring 218 ft-lb. of torque at the flywheel for starting (Table 2), tied in with a battery of 115 amp-hr. capacity, carrying the old S.A.E. rating No. 1, is equipped with a generator having a maximum output hot of 17.5 amp. A few troublesome starts on a zero morning would exhaust this battery to a point where normal operation of the vehicle with the available generator capacity would not restore this battery to a point where good operating efficiency could be expected. This engine should have been equipped with a 4H S.A.E. battery.

Probably it is unreasonable to expect the manufacturer to furnish as standard equipment the proper battery size, but it is not unreasonable to expect him to have available an optional battery size that would be satisfactory, and a battery mounting designed to carry the oversize battery in the event that the operator elected to specify it.

Table 4 – Truck Generator Characteristics – Standard Equipment

Engine #	Max. Current Amperes Cold	Max. Current Amperes Hot	S.A.E. Battery # Standard Equipment	S.A.E. Battery Rating Amp. Hr.	S.A.E. Battery Rating 300 Amp. Rate	Generator Rating Watts Cold	Generator Rating Watts Hot
a	21	18.4	1 *	90	3.1	180	153
b	21	19	1 *	90	3.0	174	158
c	20	17	2H	117	4.2	160	136
d	16.2	12.4	1 *	100	2.1	133	94
e	18	13	1 *	102	3.0	159	101
f	21	18.4	1 *	90	3.1	180	153
g	17	14.6	1 *	105	4.1	135	119
h	18	16	-	100	-	135	112
i	21.2	18	1 *	115	3.0	148	126
j	21	19	1 *	90	3.0	174	158
k	18	13	1 *	102	3.0	149	101
l	21	18.4	2 *	119	4.3	180	153
m	20	17	2H	117	4.2	160	136
n	18	13	1 *	115	3.3	149	101
o	22	16	4 *	136	5.4	176	128
p	21	19	1 *	90	3.0	174	158
q	18	15	1 *	102	3.1	149	120
r	21.2	18	2A	139	4.05	148	126
s	22	16	4 *	136	5.4	176	128
t	18	13	1 *	115	3.3	159	101
u	21.2	18	2A	139	4.05	148	126
v	18	15	1 *	102	3.1	149	120
w	25	22	2 *	119	4.3	200	176
x	21	18.4	3 *	136	5.4	180	153
y	18	15	1 *	102	3.1	149	120
z	20	17	3H	133	5.3	160	136
A	21	17.5	1 *	115	3.3	147	123
B	22	16	5 *	153	6.6	176	128
C	20	17	3H	133	5.3	160	136
D	30	30	4H	150	6.4	240	240
E	28	-	4 *	136	5.4	224	-
F	21	17.5	1 *	115	3.3	141	123
G	18	9	4 *	152	6.1	149	67
H	30	30	4H	150	6.4	240	240
I	25	22	2 *	119	4.3	200	176
J	21	17.5	1 *	115	3.3	147	123
K	18	9	4 *	152	6.1	149	65
L	28	-	3 *	136	5.4	224	-

*Old S.A.E. Rating

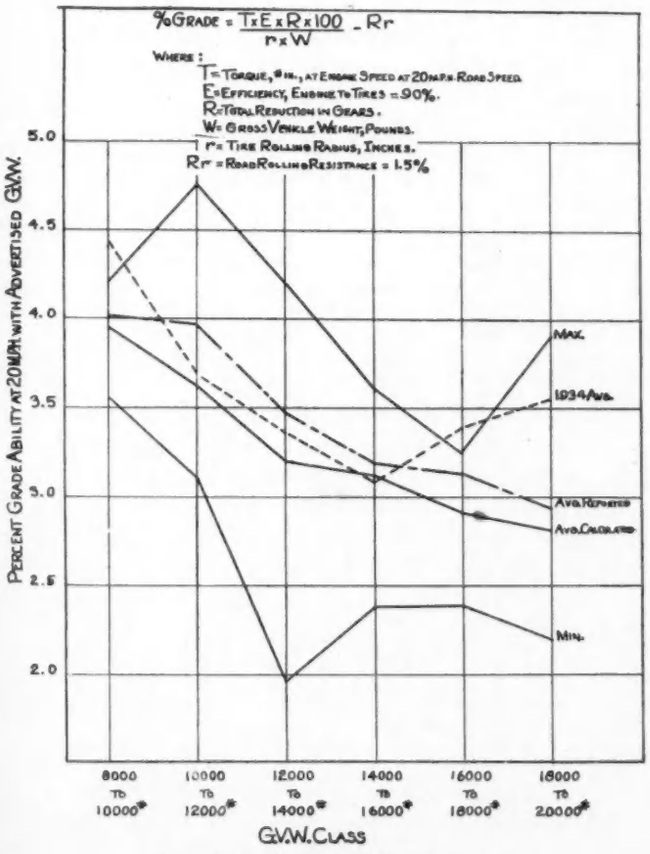


Fig. 9 – Grade-Ability Comparisons

Excerpts from Dec. 16, 1936, Letter Received from Delco-Remy Division, Anderson, Ind.

In discussing the results shown by this survey, it is probably best to consider each item of the electrical equipment separately in so far as possible.

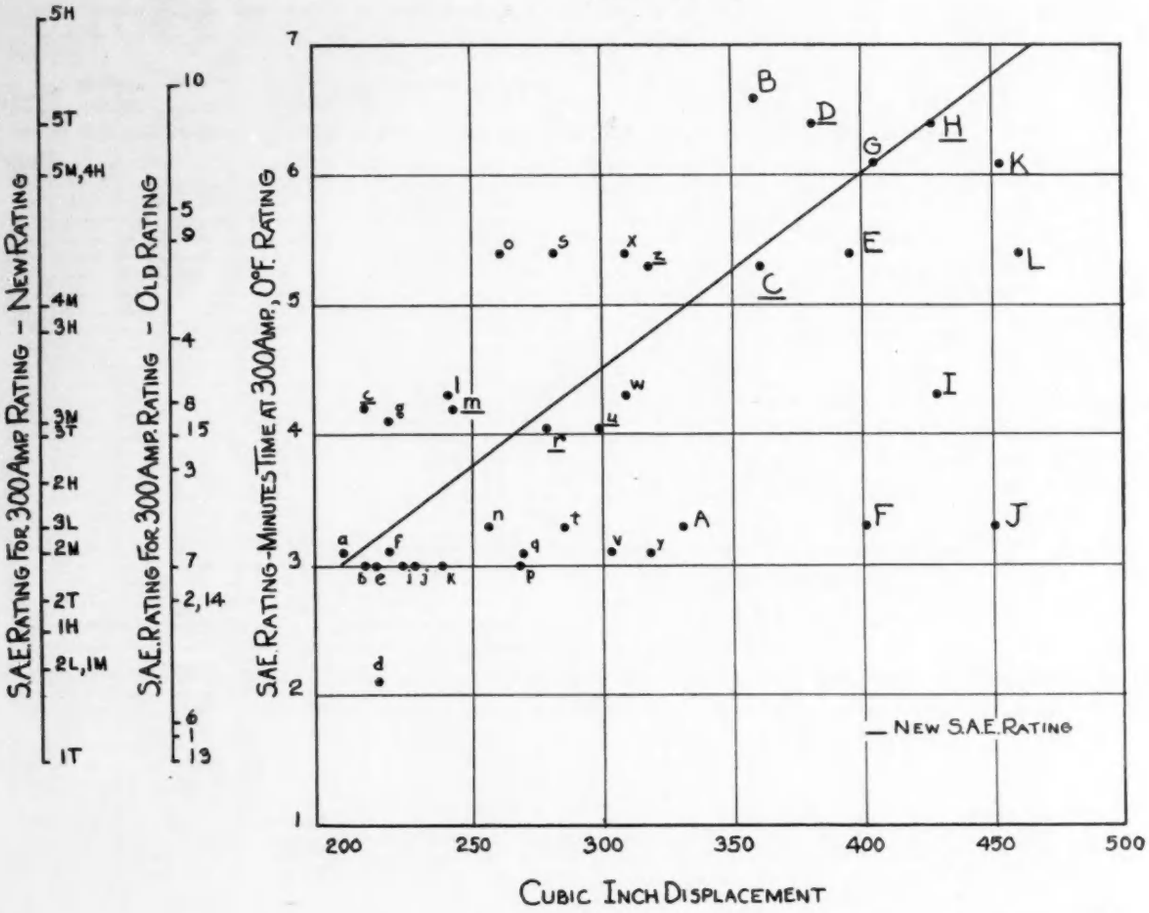


Fig. 10 – S.A.E. Rated Batteries Used for Various Displacement Engines

Generators

It is believed that truck generator requirements may be divided into the following broad classifications:

(1) *General city use.* – This type of service requires virtually a so-called standard-equipment generator as far as the electrical performance is concerned. It may be advisable to provide more rugged construction in some instances.

(2) *Door-to-door delivery.* – This application covers milk delivery, bakery trucks, and so on. The generator capacity requirements for this service are about the same as those required for item (1), but the output must be obtained at much lower speeds. Experience indicates that an output of 7 to 10 amp. should be obtained with the engine idling.

(3) *Cross-country trucking units, including tractor-trailers.* – Due to the use of marker lights to meet State requirements, fog lights, heaters, and so on, together with the requirement that operation is often in all-night service, the generator should have about 25-amp. output for 6-volt applications and 15-amp. output for 12-volt applications.

A chart has been prepared showing generator maximum output in watts against engine displacement in cubic inches. This chart shows practically no relation between engine displacement and generator capacity. This condition more or less checks with what was said previously with reference to generator applications, that is, that small generators might be satisfactory for applications on large engines where the operation did not require much electrical load.

Generator outputs as a whole, however, have been stepped up due mostly to the adoption of ventilation which has permitted higher outputs with the same size equipment. In some instances this arrangement has required the use of voltage regulation to prevent the higher-output generators from over-charging the battery or building up too high a voltage for the lights or ignition circuits for long life of lamp bulbs or ignition contacts.

Partly as a result of ventilation of the generator equipment and of the provision of low-priced regulator equipment, accessory generators of higher capacity are available for practically all engines. It is obvious, however, in view of the extra cost and the keen price competition in truck sales that this equipment will not be furnished unless specified by the operator.

Starting Motors

The starting-motor data were studied from the following angles:

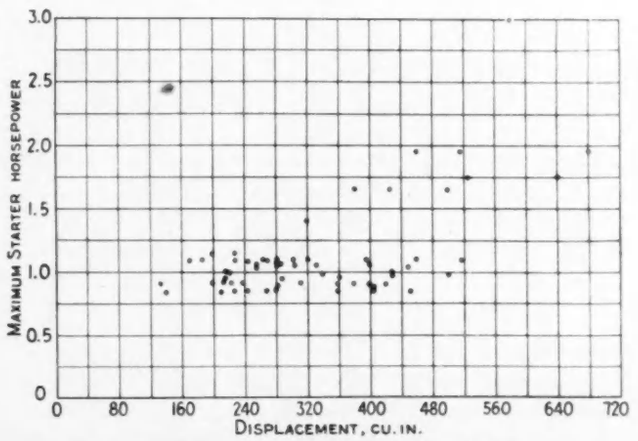


Fig. 11 – Starter Horsepower Vs. Engine Displacement

The starter stall torque at the crankshaft was plotted versus engine displacement. The starter torque figures were obtained by taking the stall torque values for each particular motor, and multiplying them by the gear reduction between the starter and crankshaft. Examination of the chart indicated that a fairly uniform relation exists between engine displacement and the break-away ability of the starter.

In order to gage cranking ability better, another chart was made up plotting maximum starter horsepower against engine displacement. (Fig. 11.) A study of this chart indicates that some of the larger engines probably do not have adequate starter capacity. However, experience indicates that other items, such as carburetion, manifolding, battery capacity, oil viscosity, and so on, have so much to do with the ability of an engine to crank at certain speeds, and to fire and run even when cranked at reasonable speeds, that it is difficult to lay down any general rules.

In this connection, it is not believed that the truck operators have adopted light winter oils to the extent that this practice has been accepted for passenger-car winter use. Fig. 12

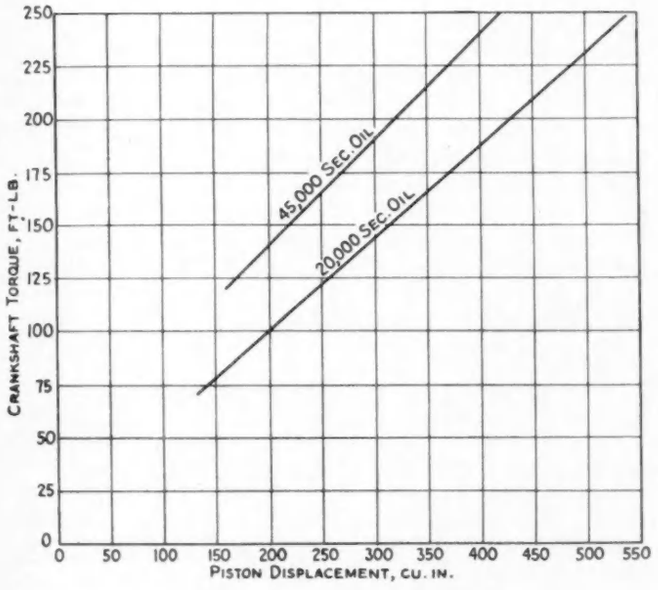


Fig. 12 – Cranking Torque Vs. Piston Displacement for Different Viscosity Oils at 0 Deg. Fahr.

shows the relation between crankshaft torque and engine displacement for two different viscosity oils at 0 deg. fahr. Taking a specific example, the 300 cu. in. engine requires about 140 ft-lb. torque to crank with 20,000-sec. oil at 0 deg. fahr. The same engine with 45,000-sec. oil requires approximately 190 ft-lb. This difference in cranking requirements could result easily in a starter application being very satisfactory with the low-viscosity oil, whereas it might not crank at all with the heavier oil. It is realized that the use of lighter oil may increase oil consumption. This increase may not be such a problem in passenger-car operation where cars are driven less, as a rule, during the winter months and, in most cases, the driving is for short distances with cooler engines. With trucking operations the service is probably as severe in the winter as in the summer, and the temperatures more nearly approach summer temperatures, so that the oil consumption might be considerably higher.

Batteries

In general, heavier batteries should be provided for the heavier engines in order that sufficient power will be available for cranking. It is believed that the new S.A.E. standard which lists the minutes at which a battery will discharge at 300 amp. rate at 0 deg. fahr., before dropping to 1 volt per cell, best measures its cranking ability.

Ignition

Heavy-duty distributors are now being provided for many trucking applications. These distributors have improved bearings and hardened parts of increased section for the automatic-advance mechanism.

Improvements in coils cannot be traced so much to radical changes in design as to gradual improvement in manufacturing technique and minor design changes resulting from increased knowledge of requirements for this type of service.

System Voltage

6-volt systems are in general use, although 12-volt accessory equipment is usually available. The use of 12 volts is generally dictated by the following requirements:

(1) *High generator capacity.* – Although no definite breaking-off point exists, it is believed that, due to inefficiency of the lower voltage system when carrying heavy currents, a change should be made to 12 volts when generator capacities in excess of 225 watts are necessary.

(2) *A requirement of high-capacity starting motors for some of the larger engines.* – Here again a definite rule cannot be established due to the variation in requirements with different engines. It is believed, however, that 12-volt starters would be advisable with displacements above 400 cu. in.

Diesel Engines

Diesel engines present a much more difficult cranking problem than do the conventional gasoline types. This difficulty is due in part to the higher compressions and heavier bearings and constructions used, and in part to the requirement in most cases for a higher cranking speed for starting (125 to 200 r.p.m. as against 30 to 50 r.p.m. for the gasoline type).

Fig. 13 brings out very forcefully the increased starter requirements for Diesel engines. It will be noted that the average passenger-car starter has a maximum output of 1 hp., whereas the large Diesel motor has an output of nearly 12 hp. Due to the increased starting motor capacities, it has been necessary, especially for the larger Diesels, to go to the 24-volt motors. In some cases the entire system is arranged

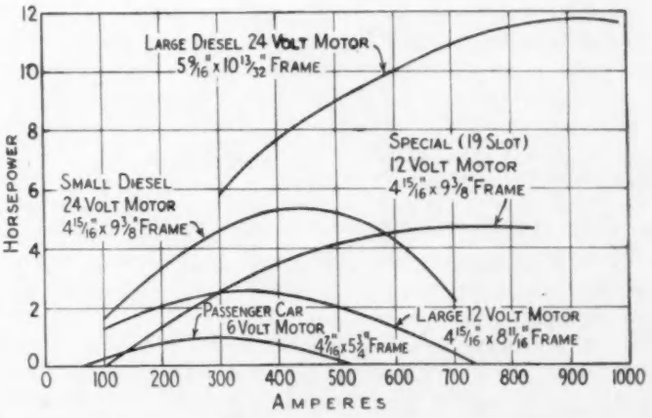


Fig. 13 – Cranking Motor Horsepower-Output Curves

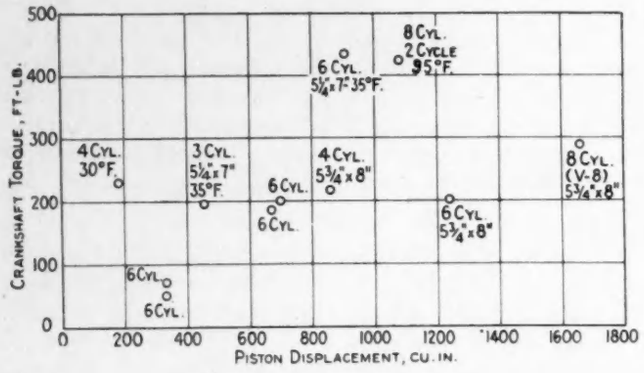


Fig. 14 – Diesel Engine Cranking Torque for Various Capacities

for 24 volts but, due to the increased cost and lack of availability of 24-volt units, some systems are provided with a 24-volt starting motor, double battery, 12-volt generator, lighting and accessory equipment, with a special starting switch which changes the battery connection from 12 to 24 volts during cranking.

Fig. 14 shows the torque required at the crankshaft for various capacity Diesel engines. These values are the results of actual tests on the engines concerned.

In general, the starter requirement is the only special one for Diesel application. Generators of virtually standard capacities are satisfactory. In view of the energy required to crank engines of this type, the importance of providing batteries of sufficient capacity is very evident. Even the smaller Diesels require batteries of 19-plate capacity and, in some instances, batteries with as many as 39 plates are either in use or being contemplated for use.

Section 3

Semi-Trailer Coupling Heights

Recognizing that interchangeability as between tractors and trailers constitutes one of the principal advantages of the tractor-semi-trailer and realizing that numerous difficulties have prevented a degree of interchangeability which would develop semi-trailer transport to its full potentialities, your committee has undertaken a study of just one phase of this problem which, if a solution were found, would facilitate materially the practical interchange of semi-trailers. This phase is that of fifth-wheel coupling heights.

Most persons are familiar with the process of semi-trailer coupling, wherein the tractor is coupled with the semi-trailer, which is parked with its front end supported by retractable support wheels, by being backed into it so that its ramping engages the underside of the trailer, raising it to the level of the turntable, while the kingpin is guided into the slot by the bifurcated legs, eventually being locked in place by the tripping of the latch.

Some Interchangeability Exists

Several different makes of fifth wheel are now in general use which are not only interchangeable with each other, but which are adaptable readily to non-standard types. The chief differences among conventional fifth wheels are to be found in the diameters of kingpins and the size and location of their grooves. Although trailer makers might, to their mutual profit, get closer together in the matter of standardizing fifth-wheel kingpins and although at a later date the good offices of the Society of Automotive Engineers might well be sought in this way; nevertheless, as things are today, practical interchangeability exists in a large measure. Real difficulty is encountered when the attempt to interchange is balked by too much disparity in fifth-wheel heights. For example, suppose a given tractor is desired for use with two different makes, types, or ownership of semi-trailers. Presumably one type of trailer has been purchased especially to go with the tractor; but the other normally operates with a different tractor. Tire sizes and frame heights may be different so that, upon attempting to couple with the second type of trailer, the plate is either too low so that, if the tractor ramp gets under it at all, the fifth wheel must assume an awkward slant; or too high so that, when the coupling is completed, the trailer support wheels are not raised from the ground sufficiently to permit them to be wound out of the way.

History of Semi-Trailer Standardization

To discover what might be done to ameliorate these difficulties, the Subcommittee on Motor-Vehicle Design and Operation of the Transportation and Maintenance Activity Committee in 1935, undertook a study of semi-trailer coupling heights. This is not the first time the Society has wrestled with this problem of semi-trailer interchangeability. As long ago as 1919, some work was started along this line without any definite accomplishment. Then, in 1928, at the suggestion of G. C. Woodruff, of the New York Central Railroad, the general subject of fifth-wheel standardization was referred to the Motor Truck Division of the Standards Committee and to the Transportation and Maintenance Activity Committee in 1929. In all, seven tumultuous meetings were held by the Society, involving the Motorcoach and Motor Truck Activity Committee, the Committee on Automotive Transportation of the A.P.I., and various manufacturers, all of which resulted in a complete stalemate.

Outlook for Standardization

Nevertheless your committee has persisted in the belief that by more clearly defining the immediate objective as the standardization of coupling heights only and in the light of the considerable progress which has been made in the industry toward de facto standardization of the conventional semi-automatic fifth wheel, there was still hope of tangible accomplishment. This study has demonstrated, at least to the satisfaction of the committee, that:

(1) Standardization of semi-trailer coupling heights is possible and feasible.

(2) Such standardization can be adopted by the industry without material change in existing designs and manufacturing practice.

(3) A standard of this type would be extremely simple.

(4) It will accomplish, in so far as coupling heights are concerned, a degree of interchangeability that is sufficient commercially to fulfill practical requirements.

Defining the Dimensions

As a first step in the study definition of the dimension or dimensions to be the subject of standardization was sought. A semi-trailer coupling consists essentially of the two parts of a fifth wheel, the upper part being a flat plate on the bottom of the front part of the semi-trailer and carrying the kingpin and the lower part being the turntable carrying the latch. There are three heights to be considered:

(1) The height of the upper surface of the lower half

when the tractor is detached, hereinafter called the tractor height.

(2) The height of the lower surface of the upper half when the trailer is detached and resting on its support wheels, hereinafter called the trailer height.

(3) The height of both of these surfaces when the tractor is coupled to the trailer, hereinafter called the coupled height.

Of these, the first is the highest, the second, the lowest and the third is in between.

Certain limitations more or less dictate the height of each. Starting with the coupled height, this distance must be great enough to provide proper clearance over the tires of the tractor to allow for maximum spring deflection plus an additional amount to care for sidewise and fore-and-aft rocking. Obviously, too, this height will vary according to the spring deflection under load, so that the height with an empty trailer will be from 3 to 4 in. above that with a load or when negotiating a road obstruction that will cause the springs to bottom. The tractor height will be as much higher than the coupled height as the spring and tire deflection from no fifth-wheel load to the loaded height. This height will vary according to the spring deflection, tire size and inflation pressure and, to a very small extent, to the distance from the axle center to the kingpin center. The trailer height must be sufficient to permit the ramp on the tractor to come below the lower surface of the upper half and yet it must not be so high as the tractor height, otherwise the support wheels will not be raised clear when coupling. It will be subject to an appreciable variation according to the load and its consequent effect upon the spring and tire deflection on the trailer.

Influence of Coupling Heights

Most semi-trailer support-wheel mechanisms are adjustable to adapt the trailer height to the tractor height of a particular tractor but, the moment a tractor with a sufficiently different height appears, the trailer height is in danger of failing to jibe. Trailer manufacturers have thus had to choose between two courses: to provide a quickly operable adjustment of support wheel height or to depend upon chassis producers to provide fifth-wheel and ramp equipment having sufficient lift to afford adequate tolerance of variations in tractor height.

That the latter course has not fully satisfied the needs of the situation is evidenced by the very fact that truck operators' complaints have three times spurred the Society of Automotive Engineers to efforts to ameliorate the difficulties that have ensued. This dissatisfaction is accounted for by the fact that the total rise of the ramp is limited by the height of the frame of the tractor on the one hand and by the tractor height on the other. Some tractor manufacturers have made the most of these possibilities by using extension ramps and even beveled frames. Trailer manufacturers have also provided beveled lips at the front. So far, the quick-adjustable support-wheel mechanism has not appeared. Certain full-automatic trailers employ a self-adjusting support-wheel arrangement but, unfortunately, this arrangement merely sets the support wheels at the proper height for the tractor from which the trailer is detached and cannot be adjusted quickly to allow for one of different fifth-wheel height. Of course, the problem would be simplified enormously if all tractor tires were of the same diameter but, unfortunately, each change in cross-section as well as each variation in rim diameter yields a different total tire diameter. It is essential on the one hand that the coupled height shall provide ample tire clearance in all cases, and it is also highly desirable that this height shall be no more than absolutely necessary in the interests of low loading height and low center of gravity.

Obviously from the foregoing, it will be apparent that the coupling height simply cannot be reduced to a uniform standard or series of standards, but it must be allowed to come as it will according to the load, spring deflection, tire deflection, and tire size. There seems to be considerably more possibility of reaching some uniformity of practice with respect to tractor height and trailer height. Tractor height is established simply on a basis of distance above the height of the tires; but, with reference to the ground, this basis will produce results that vary tremendously through the entire range of tire sizes in common use. Trailer height is controlled more easily since the support wheels can be designed and adjusted to any given height desired.

Tire Height the Basis

An approach was made, therefore, from the standpoint of standing heights of various tire sizes, necessary clearances, and prevailing spring deflection.

Table 5 – Tire Heights of Truck Tires

Tire Size		Outer Diameter	Loaded Radius	Tire Hgt. $\frac{D}{2}+R$
Balloon	High-Pres.			
5.50-20		32.1	15.19	31.24
	30 x 5-6	32.2	15.25	31.35
6.00-20		32.6	15.42	31.72
	30 x 5-8	32.5	15.71	31.96
	32 x 6-8	33.4	16.02	32.72
6.50-20		33.8	15.98	32.88
	32 x 6-10	34.5	16.72	33.97
7.00-20		35.0	16.49	33.99
7.50-20		35.8	16.90	34.80
	34 x 5	36.6	17.70	36.0
	34 x 7	36.6	17.76	36.06
8.25-20		37.3	17.70	36.35
9.00-20		38.7	18.20	37.55
	36 x 6	38.6	18.70	38.0
	36 x 8	38.6	18.70	38.0
8.25-22		39.4	18.72	38.42
7.50-24		39.9	18.84	38.79
9.75-20		40.2	18.87	38.97
9.00-22		40.9	19.16	39.61
	38 x 7	40.6	19.67	39.97
10.50-20		41.4	19.55	40.25
8.25-24		41.4	19.64	40.34
	38 x 9	41.2	19.88	40.48
9.75-22		42.1	19.84	40.89
9.00-24		42.9	20.12	41.57
11.25-20		43.1	20.23	41.78
	40 x 8	42.7	20.71	42.06
10.50-22		43.5	20.50	42.25
11.25-22		45.1	21.07	42.62
9.75-24		44.2	20.82	42.92
	40 x 10	43.9	21.17	43.12
12.00-20		44.9	21.21	43.66
10.50-24		45.6	21.44	44.24
	42 x 9	45.3	21.86	44.51
11.25-24		47.1	22.13	45.68
	44 x 10	48.0	23.14	47.14
12.00-24		48.8	22.95	47.35

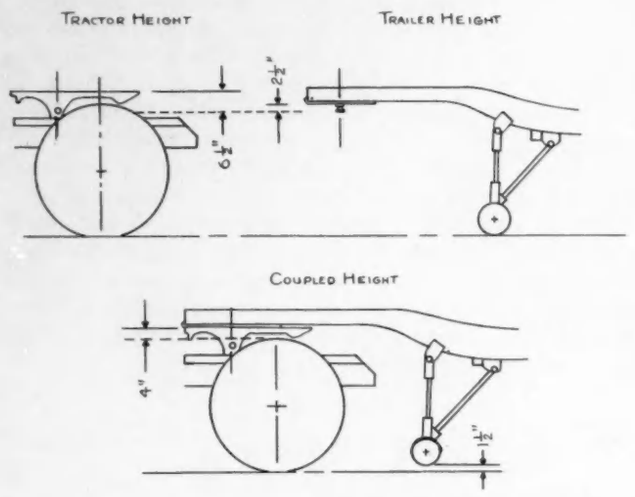


Fig. 15 – Ideal Tractor Height, Trailer Height, and Coupled Height

Starting with a given tire size, its outer diameter is reduced by the amount of flattening under load so that the actual tire height is the sum of the loaded radius plus half the outer diameter.

Clearance over the tires with the springs fully loaded must be sufficient to allow for tire chains (2 in.) plus an additional amount for overload or impact deflection of springs to full-bottomed position (1½ in.) or a total of 3½ in. Usual semi-trailer construction involves outriggers over the tractor tires which have their lower surfaces somewhat elevated above the bottom level of the main frame members, either by being made of shallower sections or by being tapered. At least 1 in. of rise may be expected from this source, which will bring the bottom of the frame side-member 2½ in. above the top of the tires. This value represents the absolute minimum coupled height for this size of tire, the maximum being as much greater than this as spring deflection makes it, say 4 in.

Obviously, since each diameter and cross-section of tire will change these dimensions, a single standard of coupling height with reference to the ground is impracticable, so any standard to be of use must be in terms of height above the tires. Table 5 shows the tire height of truck sizes of both balloon and high-pressure tires. Turning, accordingly, to trailer height – the height of the underside of the upper half of the fifth wheel when the trailer is parked – we find that the requirements to be satisfied are that the support wheels, when in the lowered position, must clear the ground by approximately 1½ in. when the trailer is coupled to the tractor and with maximum fifth-wheel load imposed. This height, as we have seen, will be about 1½ in. above the minimum and bottomed coupled height. As the two dimensions cancel, it may be stated that the trailer height should be equal to the bottomed coupled height.

In such case the support wheels would have no clearance above the ground – a desirable restraint against excessive overloading – but, under normal full load, the process of coupling will raise the trailer 1½ in. higher than its parked height.

This consideration brings us to tractor height – the height of the upper surface of the fifth-wheel lower half when the trailer is detached from the tractor. Ideally, this height would be equal to the bottomed coupled height plus the amount of the spring deflection (4 in.), or 4 in. higher than the trailer height – the bottom surface of the upper half when the trailer is parked.

Of these three dimensions, the trailer height is the only one that can be maintained accurately, depending, as it does, upon the adjustment of the support-wheel mechanism. As we have seen, this works out to 2½ in. minimum above the height of the tires. Working from this starting point, the tractor height must be equal to this height plus the full deflection of the tractor rear springs (4 in.) or 6½ in. above the tires. The difference in height between the lower and upper halves is cared for in the coupling operation, of course, by the ramp action, due to the tilting of the turntable and its ex-

Table 6 – Interchangeability Among Tractors and Trailers

Tire Size		A Height of Tire	Tractor Height A – 6½"	Trailer Height A – 2½"	Interchangeable on Tractor with the Trailer sizes:	
Balloon	High-Pres.				From	to (incl)
5.50-20		31.24	37.74	33.74		
	30 x 5 (6)	31.35	37.85	33.85	5.50-20	30x5 (6)
6.00-20		31.72	38 22	34.22	5.50-20	6.00-20
	30 x 5 (8)	31.96	38.46	34.46	5.50-20	30x5 (8)
	32 x 6 (8)	32.72	39.42	35.42	5.50-20	32x6 (8)
6.50-20		32.98	39.38	35.38	5.50-20	6.50-20
	32 x 6 (10)	33.97	40.47	36.47	5.50-20	32x6 (10)
7.00-20		33.99	40.49	36.49	5.50-20	7.00-20
7.50-20		34.80	41.30	37.30	5.50-20	7.50-20
	34 x 5	36.00	42.50	38.50	30 x 5(8)	34x5
	34 x 7	36.06	42.56	38.56	32 x 6(8)	34x7
8.25-20		36.35	42.85	38.85	32 x 6(8)	8.25-20
9.00-20		37.55	44.05	40.05	32 x 6(10)	9.00-20
	36 x 6	38.00	44.50	40.50	7.00-20	36x8
	36 x 8	38.00	44.50	40.50	7.00-20	36x8
8.25-22		38.42	44.92	40.92	7.50-20	8.25-22
7.50-24		38.79	45.29	41.29	7.50-20	7.50-24
9.75-20		38.97	45.47	41.47	7.50-20	9.75-20
9.00-22		39.61	46.11	42.11	34 x 5	9.00-22
	38 x 7	39.97	46.47	42.47	34 x 5	38x7
10.50-20		40.25	46.75	42.75	34 x 5	10.50-20
8.25-24		40.34	46.84	42.84	8.25-20	8.25-24
	38 x 9	40.48	46.98	42.98	8.25-20	38x9
9.75-22		40.89	47.39	43.39	9.00-20	9.75-22
9.00-24		41.57	48.07	44.07	9.00-20	9.00-24
11.25-20		41.78	48.28	44.28	36 x 6	11.25-20
	40 x 8	42.06	48.56	44.56	36 x 6	40x8
10.50-22		42.25	48.75	44.75	8.25-22	10.50-22
11.25-22		42.62	49.12	45.12	7.50-24	11.25-22
9.75-24		42.92	49.42	45.42	9.75-20	9.75-24
	40 x 10	43.12	49.62	45.62	9.00-22	40x10
12.00-20		43.66	50.16	46.16	9.00-22	12.00-20
10.50-24		44.24	50.74	46.74	10.50-20	10.50-24
	42 x 9	44.51	51.01	47.01	8.25-24	42x9
11.25-24		45.68	52.18	48.18	9.00-24	11.25-24
	44 x 10	47.14	53.64	49.64	40 x 10	44x10
12.00-24		47.35	54.85	49.85	12.00-20	12.00-24

tension legs, plus any ramping of the tractor deck, beveling of the tractor frame, and the bevel lip of the semi-trailer plate. The setting of the tractor turntable at the height desired is effected by mounting the fifth-wheel pedestals with filler blocks between them and the frame side-members or on cross-bolsters of the proper thickness.

Ideally, therefore, our three heights would be as shown in Fig. 15, in terms of inches above the height of the tires.

Interchangeability Afforded

To determine the extent of interchangeability afforded, the tolerance of variation in height permitted by what may be referred to as ramp rise must be considered. In the ideal set-up as in Fig. 15, a ramp rise of 4 in. would suffice. As a matter of fact, prevailing fifth-wheel types have a height of about 8 in. from pedestal base to top of turntable. When disconnected, the turntable slants to rearward so that it, with its extension legs, forms a ramp having a minimum of 8 in. rise.

To the extent that it is raised above the frame, its greater consequent inclination, extension ramping on the tractor deck, beveling of the tractor frame and the beveled lip on the front of the semi-trailer, extend this ramping so that not less than 10 in. and, in some cases considerably more, can be relied upon as the available ramp rise. This value is 6 in. or more in excess of ideal requirements, which means a tolerance with respect to trailer height of 6 in. on the minus side. The reason that this tolerance is all on the minus side, of course, is the requirement that coupled height shall be as low as proper tire clearance will permit.

Referring to Table 6 then, tractor and trailer heights have been charted in such manner as to indicate the degree of interchangeability among tractors and trailers thus afforded. In this relationship the theoretical interchangeability has been reduced somewhat in order to care for differences in tire inflation, uneven load distribution, and unevenness of road surface by cutting off the extreme inch of tolerance at each end.

A Proposal

From this discussion it will be seen that the only practicable form of standardization of coupling heights seems to be on a basis of height above the tire in each case, irrespective of the size of tire. Accordingly the standard proposed is as follows:

The height of the underside of the fifth-wheel upper half on semi-trailers shall be equal to the height of the tires[1] above the road surface plus 2½ in., measured on the transverse centerline of the kingpin.

The height of the upper surface of the fifth-wheel lower half on tractor-trucks shall be equal to the height of the tires above the road surface plus 6½ in., measured on the transverse centerline of the kingpin socket, with the turntable parallel with the frame. (See Fig. 15.)

[1] The height of tire shall be the sum of the loaded radius plus one-half the total diameter.

Budget Control in A Big Car Plant

MANY do not go far enough into the requirement of proper budgeting and effective control to accomplish the desired result – producing any given volume of production up to plant capacity at a certain given rate of expense.

To carry out an effective method of expense control in the manufacturing division in automotive plants, it is first necessary to translate the manufacturing program as it applies to the various departments into estimates of cost and then effect a positive control.

What is meant by a positive control is catching the expense before the money is spent; any other method as applied to budget work could naturally be termed "remote control."

There are undoubtedly several causes that determine the success or failure of a budget system to properly control expense in any specific plant. The more important causes might be enumerated as follows:

1. Proper analysis and methods used to establish budgets.
2. Proper administration of budget control.
3. Whole-hearted support by the management.
4. Cooperation of all department heads.

Of the four listed causes the first and second are undoubtedly the most important. The reason is obvious. Unless great care is taken when establishing the budgets and administering the control it would be difficult to obtain the support of the management and the cooperation of the department heads. Therefore, the responsibility for the lack of this support and cooperation rests to a great extent upon the budget supervisor. In many cases, it is apparently the result of the budget supervisor's failure to thoroughly understand the management's problems and the important function the budget should perform in the exercising of executive control.

Budgets properly established and supervised have a very definite place in the science of modern business management. The management's final objective in business is net profit. He realizes that all of his efforts must be directed toward this objective. Profits, or the lack of profits, is the result of numerous causes. So far as the management is concerned some of the causes are non-controllable. On the other hand, items such as the amount of money spent to produce a product are controlled by the management. The management to a large extent makes and controls the profits through the activities of its organization.

It is necessary that the budgets represent the ideal condition so far as is humanly possible; they should not penalize or favor the department regardless of any fluctuations in volumes of production. At the same time they should convey the desired psychological effect to the department head. The budgets should tie into the factory burden rate in such a manner as to afford ready analysis of the factors causing any unabsorbed burden during the month. However, it is impracticable to use the factory burden rates as a budget. For these reasons an efficient budget set-up is necessarily complex; but at the same time it must be flexible.

By establishing a relationship between our budgets and the factors of time and production activity we obtain the best results. The relationship of our budgets to the factor of time is represented by a non-varying amount of money based on one month and is used as a control on those expenses which are influenced by time and not by plant activity. These are called non-variable budgets. Variable rates expressed in terms of dollars of allowance per $100 of direct labor are used.

Excerpts from the paper: "Budgeting Expense and Cost of Handling Materials in Automotive Plants," presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 15, 1937, by George Miller, budget supervisor, Chrysler and Chrysler-Kercheval Plants of Chrysler Corp.

Relative Knocking Characteristics of Motor Fuels in Service

By John M. Campbell, Wheeler G. Lovell, and T. A. Boyd
General Motors Research Corp.

IN this investigation some of the underlying principles affecting the knocking characteristics of motor fuels in service have been studied. Briefly, the experiments indicate that the relative knocking characteristics of certain cracked gasolines with respect to straight-run gasolines may be affected by the relationship among spark timing, engine speed, and mixture ratio. From the standpoint of engine design the most advantageous combination of these variables in a given engine varies according to the nature of the fuel used.

It is suggested that some of the anomalous characteristics of benzol blends are the result of vaporization phenomena in the induction system, as also are certain "depreciation" effects which occasionally have been observed among straight-run gasolines containing tetraethyl lead.

VARIATIONS in the relative knocking tendencies of different types of fuels from one car to another have become recognized widely in recent years. As a result of these variations the terms "depreciation" and "appreciation" have been used to express the idea that a certain fuel appears worse or better in service than some standard fuel which it matched in the laboratory test engine. By definition the terms "depreciation" and "appreciation" are relative to certain predetermined standard fuels and engine conditions. Thus the question of whether a given fuel "depreciates" or "appreciates" in a given engine depends upon the point of reference that is taken. Although the authors recognize the conventional standards of reference fuels and engine conditions with respect to which "depreciation" is measured commonly, in the discussion that follows an attempt is made to make clear up how, with different standard fuels and with different standard engine conditions, the fuels which now appear to "depreciate" would then appear to "appreciate."

Fuel Factors Influencing Depreciation. – It is obvious at the outset that the method of rating fuels for knock, or for determining their relative antiknock values, depends largely upon the use to which they are to be put. Fundamentally, the use to which fuels are to be put is to produce power without knocking in an automobile engine. It is possible to think of rating fuels in terms of the brake mean effective pressure that it is possible to produce with them, and this would seem to be a fairly fundamental method and free from the use of a standard reference fuel. Such a method, however, is subject to a considerable disadvantage in that, for a definite example, fuel *X* may be better than fuel *Z* at one speed and produce more knockless b.m.e.p. but, at another speed, fuel *Z* may be better than fuel *X* in this respect. Which of the two depreciates or gets relatively inferior depends entirely upon which one is considered the standard and what speed one uses as a basis in making the comparison. For some purposes it may be more desirable to have a high b.m.e.p. at high speeds and, for other purposes, a high b.m.e.p. at low speeds, so that the relative value of a fuel in producing knockless b.m.e.p. depends upon the use to which the fuel is to be put.

As a concrete example, it is possible to design a car so that it will knock at high speed, or at low speed; in some engines it is possible to control the speed at which knock appears on a given fuel simply by adjustment of the automatic spark advance. If one makes a car with a low-speed knock, then some highly cracked fuels appear better than straight-run fuels of the same octane number. If one changes the car so that it knocks at high speed, then the opposite is true and the

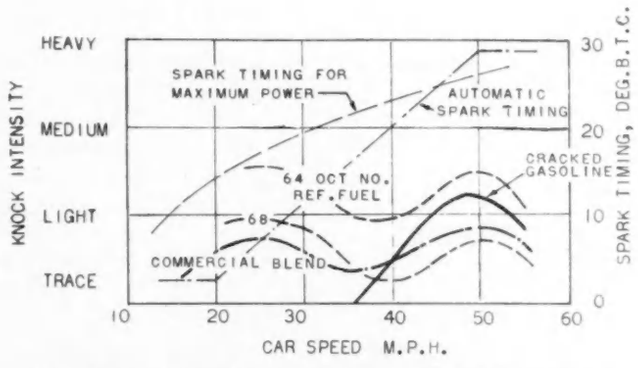


Fig. 1 – Knocking Characteristics of Two Different Gasolines in a 1934 Production Car

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 11, 1937.]

straight-run reference fuels appear good with respect to the highly cracked fuels. The question is, which is the better procedure to follow?

The choice of a procedure that will eliminate the "depreciation" depends upon the fuel that is used as a standard of reference. For example, if one uses the straight-run fuel as a reference, as has been done in the past, then the car with the low-speed knock will reduce the "depreciation" of some highly cracked fuels. If some highly cracked fuel is used as the standard of comparison, then the car with the high-speed knock will reduce the apparent "depreciation" of the straight-run fuels. Finally, if one rates two fuels that have similar sensitivities to operating conditions, then they would be expected always to rate the same both in the laboratory and on the road and irrespective of whether the cars knock at high or low speeds; so there is no depreciation as far as these fuels are concerned. Thus, the engine design which will be "ideal" with respect to "depreciation" depends upon the choice of reference fuels. If one builds an engine to eliminate all "depreciation" in the commonly accepted use of the term, that is, with respect to the straight-run reference fuels now in use, then

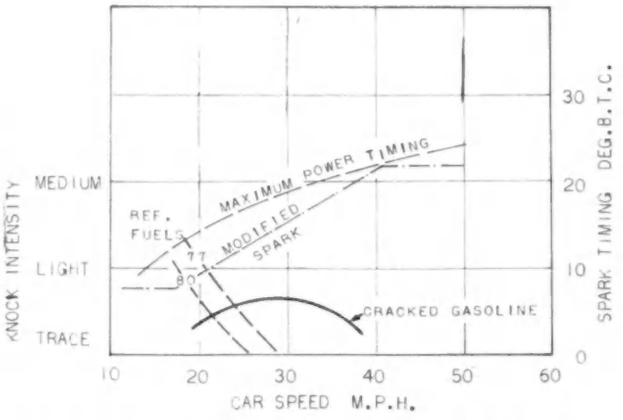


Fig. 2 – Knocking Characteristics of Car Represented in Fig. 1 with Modified Automatic Spark Timing

that means building an engine to operate best on a purely straight-run type of fuel which the car owner seldom, if ever, uses. If one builds an engine to operate best (that is, with the greatest knockless b.m.e.p. over a speed range) on a very highly cracked fuel, which the owner seldom has an opportunity to use, then some other commercial fuels do not appear well, or they "depreciate" in terms of the cracked fuel. If, on the other hand, one uses as a standard a fuel that is an average of commercial gasolines, then one can build an engine for this fuel and it will be found that, in so far as engine speed affects the knock ratings, some fuels "depreciate" a little and some "appreciate" a little and the spread will not be very great. This method is substantially what is now being used in practice; that is to say, commercial gasolines rather than straight-run reference fuels are used in practical engine development work.

Effect of Engine Speed. – In the foregoing discussion an attempt has been made to point out how the mental conception of "depreciation" that is formed as a result of road tests depends largely upon the reference fuels that are used in the road tests and that the "ideal" in engine design, which will produce the least depreciation, changes with a change in

[1] See S.A.E. TRANSACTIONS, May, 1935, pp. 165-179; "C.F.R. Committee Report on 1934 Detonation Road Tests," by C. B. Veal.

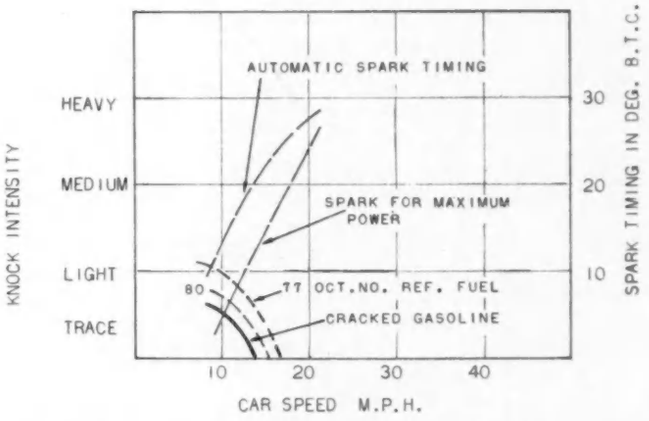


Fig. 3 – Valve-In-Head Engine with Low-Speed Knock Characteristic – Cracked Gasoline Equivalent to 82-83 Octane Number in Reference Fuel

reference fuels. The following discussion describes some actual tests indicating that the apparent "depreciation" that is observed with a particular combination of reference fuels and test fuels may be controlled by suitable control of the engine speed at which the fuels knock.

This method of control was first suggested in a report of the cooperative fuel tests made at Uniontown in 1934[1], in which it was shown that for one of the 100 per cent cracked gasolines there might be a relation between the road rating and the car speed at which the rating was made.

Continuing this investigation with some of the same type of cracked gasoline for which this relationship was indicated by the 1934 Uniontown tests, the authors have obtained some data which show how the apparent rating of a cracked gasoline may be changed in a given car engine by changing the speed for maximum knock through alterations in the automatic spark timing. The data as they were obtained from a 1934 production car having an L-head engine are presented in Fig. 1. With the standard spark curve the cracked fuel showed a high-speed knock that was absent completely at speeds below 35 m.p.h. However, by retarding the spark at high speeds, this high-speed knock was eliminated. Then, when the entire spark curve was advanced from an initial setting of 2 deg. to an initial setting of 8 deg. in order to produce knocking, maximum knock on the cracked fuel occurred between 25 and 30 m.p.h. and the apparent road knock rating was raised from an octane number of 66 with the standard spark to an octane number in the vicinity of 75-80 with the modified spark timing as shown by Fig. 2.

A summary of the octane ratings pertaining to these fuels follows:

Fuel	C.F.R. Research	A.S.T.M.	Road, standard spark	Road, modified spark
100 per cent cracked gasoline	79.9	70.7	66	80 (approximate)
Commercial blend	71.5	69.3	68	not rated

These road data, where the speed of maximum knock varies with the fuel, are interpreted in accordance with the standard Uniontown procedure. If another procedure is used, the interpretation of the data might be somewhat different.

Fig. 3 represents data obtained on another make of 1934 car having a valve-in-head engine. In this car maximum knock occurred at the lowest speed at which it was practicable to run, and the rating of the same cracked gasoline was estimated to be between 82 and 83 octane number.

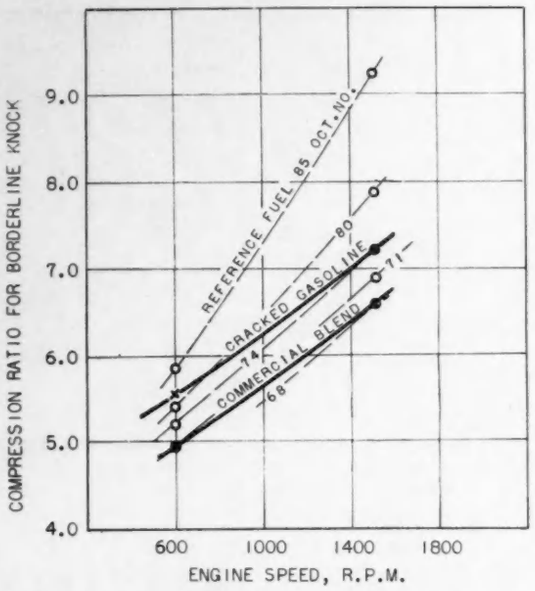


Fig. 4 – Effect of Engine Speed on Knock Ratings in a Single-Cylinder, Variable-Compression Engine – Spark Timing and Mixture Ratio for Maximum Power; Jacket. 212 Deg. Fahr.; Mixture, 70 Deg. Fahr.

In an effort to obtain some data on how speed would affect ratings in a valve-in-head engine which would be comparable to the effect of speed in the L-head engine as shown in Figs. 1 and 2, the experiments illustrated in Fig. 4 were made in a laboratory single-cylinder, variable-compression engine. These tests were made in a single-cylinder engine because it was found impracticable to advance the spark in the car engine enough to obtain maximum knock at the higher speeds. Fig. 4 shows the rate of change of compression ratio for borderline knock with increasing engine speeds. The figure shows a greater rate of change in the compression ratio for borderline knock with the reference fuels than with the cracked gasoline.

These data clearly illustrate the "depreciation" phenomenon and how it changes with the reference point. For example, at 600 r.p.m., borderline knock for the cracked gasoline was at the same compression ratio as that for an 82 octane number reference fuel (by interpolation). But the rate at which the borderline knock compression ratio increases with speed is greater for the reference fuels than for the cracked fuel. For example, at 1500 r.p.m., the cracked fuel is equal in knock to a 74 octane number reference fuel. It appears to "depreciate" the equivalent of 8 octane numbers upon changing the speed from 600 to 1500 r.p.m. On the other hand, if we take 1500 r.p.m. as a reference point, we might say that the cracked gasoline "appreciated" 8 octane numbers in changing the speed from 1500 to 600 r.p.m. or, if we think of the cracked gasoline as a standard, then the "reference" fuels "depreciated."

Thus it appears from the data in Figs. 1 to 4 that the apparent "depreciation" of cracked gasolines with respect to straight-run reference gasolines may be controlled to a certain extent by controlling the speed for maximum knock. The data in Figs. 1 and 2 indicate that this control can be accomplished by means of a suitably regulated spark-timing curve. Whether it is wholly desirable to base our thinking altogether on the use of straight-run reference fuels is a matter about which there can be some question. In this connection it might be pointed out that the passenger-car engine represented in Figs. 1 and 2 had a relatively high compression ratio and was provided with an automatic spark which was from 5 to 10 deg. retarded with respect to the spark for maximum brake torque at low speeds. Thus the spark was retarded purposely at low speeds in order to avoid excessive knocking on the average commercial type of gasoline for which the engine was intended. Fig. 1 indicates that the spark was held within production limits of borderline detonation up to about 60 m.p.h. for the standard car engine. This policy of keeping the spark retarded and at borderline detonation at low speeds and full throttle in order to avoid detonation in a high-compression engine appears to be a sound practice under some conditions.

However, Fig. 1 shows that what is a borderline timing for one fuel may be something entirely different for another fuel. Thus a borderline-detonation spark-timing curve for the cracked gasoline would produce severe knocking at low speeds with an ordinary commercial gasoline. In other words the commercial gasoline would appear to be greatly "depreciated" at low speeds. Since this particular cracked gasoline was not a representative commercial gasoline, setting of the spark timing for best operation with such a fuel would not be a wise thing to do because most commercial fuels for which the car was intended would then be "depreciated" at low speeds and knocking would become a troublesome problem with these fuels. The borderline knock curve should therefore be based on the type of commercial gasoline for which the car is intended.

Effect of Mixture Ratio. – Another factor that may affect the apparent depreciation of fuels in cars is mixture ratio. The data in Fig. 5 show that, at a carburetor setting corresponding to a mixture ratio of 14:1, which was the maximum knock setting as well as approximately the leanest mixture for maximum power, the cracked gasoline was equivalent to a reference fuel of 80 octane number. At a carburetor setting corresponding to a mixture ratio of 11:1, the same cracked gasoline was equivalent to a reference fuel of only 74.5 octane number. Thus an apparent "depreciation" of 5.5 octane numbers was brought about merely by changing the mixture ratio from 14:1 to 11:1.

This fact is of significance in connection with the 1934 Uniontown road ratings on this same type of gasoline in which it was found that, although the low-speed ratings were generally higher than the high-speed ratings, they were still appreciably below the research ratings made at 600 r.p.m. and at maximum knock mixture ratios which are in the neighborhood of 14:1. The mixture ratios in the cars observed from exhaust-gas analysis taken at the time of the road tests were from 11:1 to 12:1 and the discrepancy between the research rating and the low-speed road ratings may be accounted for reasonably on the basis of these differences in mixture ratios.

It is theoretically desirable, of course, from the standpoint of economy to have cars operate on as lean a mixture as possible, and all such advances in that direction would tend to reduce the apparent depreciation with respect to the straight-run fuels. These changes, in addition, would increase the tendency to knock beyond that prevailing with the present richer mixtures.

Effect of Fuel Volatility. – In addition to variations in knocking characteristics among different fuels caused by differences in chemical composition, there is evidence presented here that in certain cases the volatility factor may contribute in a considerable degree to the knocking characteristics of a

fuel in a passenger-car engine on the road. It is believed that this effect of volatility is particularly applicable to what hitherto has been considered the anomalous behavior of benzol blends as a result of which benzol blends almost invariably appear better in road tests than otherwise would be indicated by the conventional laboratory knock rating.

This behavior on the part of benzol blends is just what should be expected on the basis of the following analysis of the physical processes that take place in the induction system during acceleration when knock measurements are made. In this analysis it is assumed that, during acceleration at wide-open throttle, a wet air-fuel mixture is being delivered to the engine. This mixture, being wet, is not distributed equally to all cylinders, nor does any one cylinder necessarily receive the same mixture during each successive cycle. Accordingly, some charges are lean, some are rich and, when the average mixture ratio is richer than the maximum knock mixture, whatever knocking occurs would be expected to originate from charges which are on the lean side of the average. However, if the fuel contains a relatively large proportion of benzol, which is a comparatively volatile fuel boiling at 176 deg. fahr., then the lean cylinders will receive a fuel that is richer in benzol than the original fuel. Consequently, due to the added antiknock effect of this benzol, these lean charges will have less tendency to knock than they would if the fuel were a conventional gasoline of the same antiknock rating in the single-cylinder laboratory engine.

Another element in the vaporization process that would be expected to affect knocking characteristics is the tendency during the first part of the accelerating period for the less volatile fractions of the gasoline to separate on the walls of the induction system. This separation would cause the cylinders temporarily to receive a mixture rich in benzol when a benzol blend was used. These factors all would tend to cause an apparent "appreciation" of the benzol blend with respect to a conventional gasoline in a multicylinder engine on the road. Conversely, according to this picture, the addition of a large proportion of a volatile component of low antiknock rating to a gasoline of high antiknock value should result in a fuel which knocks more in road tests than would be expected from the single-cylinder test-engine test.

As a means of testing these concepts two fuels having equal knock ratings by the current A.S.T.M. method were prepared: one comprising a benzol blend in a straight-run gasoline of low knock rating, *A*-3, and the other a blend of ethyl ether in a straight-run gasoline of high knock rating, *C*-8. On the road in the car in which these fuels were compared with respect to reference fuels *A*-3 and *C*-8, the benzol blend "appreciated" about 3 octane numbers and the ether blend "depreciated" about 4 octane numbers. In terms of knock intensity, irrespective of reference fuels, where the ether blend produced a light to medium knock, the benzol blend produced no knock at all.

Vaporization of Antiknock Agents. – As stated previously, certain of the antiknock characteristics of benzol blends can be accounted for by analysis of vaporization processes taking place in the induction system. Likewise, certain characteristics which have been observed occasionally among gasolines containing lead tetraethyl can be accounted for by similar reasoning. For example, instances are known where a straight-run fuel containing lead tetraethyl unquestionably will appear several octane numbers lower in road tests than by the laboratory rating with respect to reference fuels. That this phenomenon is not associated with the more familiar "depreciation" of cracked gasolines with respect to reference gasolines is indicated by the fact that the gasoline under discussion may have identical ratings by the C.F.R. Research and Motor procedures. The explanation for this behavior seems to center around the volatility of the antiknock agent itself and its relation to the vaporization process in the induction system during the accelerating period when knock ratings are being made.

This characteristic is illustrated by the example given in the following table in which a comparison is made between a straight-run type of fuel containing lead tetraethyl which has a comparatively low vapor pressure at the temperatures existing in the induction system, with the same fuel treated with lead tetramethyl which has a vapor pressure 30 to 40 times higher than lead tetraethyl at the temperatures existing in the induction system. Both fuels had approximately the same A.S.T.M. octane number:

	Octane Number	
Rating	3.0 Cc. Lead Tetramethyl	3.0 Cc. Lead Tetraethyl
C.F.R. Motor (A.S.T.M.)	76	77
Car No. 1	78	68
Car No. 2	83	72
Car No. 3	82	74
Average car rating	81	71

In this comparison the two fuels which had nearly identical A.S.T.M. ratings differed by an average of 10 octane numbers in road tests. It is believed that this difference can be accounted for by differences in the relative volatility of lead tetramethyl and tetraethyl lead. During the first part of the accelerating period it is probable that tetraethyl lead is delayed in reaching the cylinders due to its separation on the intake manifold walls together with some of the less volatile fractions of the gasoline. Tetramethyl lead, being more volatile than tetraethyl lead, is not delayed in this way, and so its antiknock effect is felt immediately in the cylinders. At certain times the cylinders may even receive a mixture relatively rich in tetramethyl lead due to its evaporation with the lighter gasoline fractions leaving the heavy fractions on the walls of the induction system.

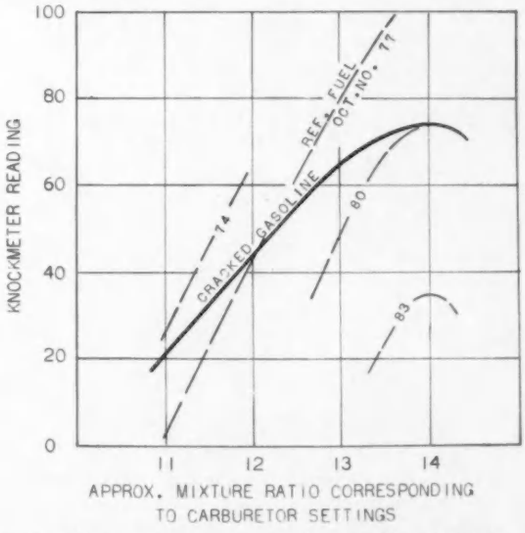


Fig. 5 – Effect of Carburetor Setting on Knock Rating of a Cracked Gasoline with Respect to Reference Fuels *A*-3 and *C*-8 – Single-Cylinder, Valve-In-Head Engine; Compression Ratio, 6.3:1; Speed, 750 R.P.M.; Spark, 30 Deg. Before Top-Center; Jacket, 212 Deg. Fahr.; No Heat on Mixture

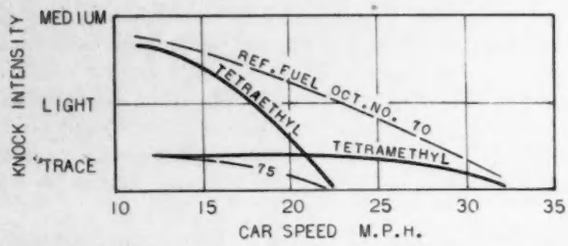


Fig. 6 – Comparison of Tetramethyl Lead and Tetraethyl Lead in a Straight-Run Gasoline – Atmospheric Temperature, 45 Deg. Fahr.; Normal Mixture Heat; Mixture Temperature 105-110 Deg. Fahr. at 20 M.P.H.; Spark Timing Adjusted for Suitable Knock

Some of the experimental facts that are in conformance with this view are as follows: first, the "appreciation" in knock rating for straight-run fuels containing tetramethyl lead and the corresponding "depreciation" of straight-run fuels containing tetraethyl lead have been observed only at low car speeds during acceleration. Second, these "appreciation" and "depreciation" effects, which are almost invariably associated, can be reduced and even eliminated by the application of sufficient heat to the fuel-air mixture.

Both of these observations are illustrated in the data shown in Figs. 6 and 7 in which a comparison was made between two straight-run fuels containing 2.06 cc. of tetramethyl lead and 3.0 cc. of tetraethyl lead, respectively, at two different settings of the mixture heat-control valve on a standard 1936 automobile. In Fig. 6, representing normal mixture heating at an atmospheric temperature of 45 deg. fahr., knock intensity as judged by ear was plotted against car speed during acceleration from 10 m.p.h. on a 7 per cent grade.

The data indicate how at 15 m.p.h. with normal mixture heating, the fuel containing tetramethyl lead knocked definitely less than the fuel containing tetraethyl lead. On the other hand, when the mixture heating was increased as shown in Fig. 7, the order in which the fuels knocked was reversed.

Of further significance is the fact that this change in mixture heating had practically no effect on the reference fuels; it made the fuel containing tetraethyl lead knock less and the fuel containing tetramethyl lead knock more. This change resulted not only in a change in the relationship between the two fuels containing lead but also in a complete change in the road ratings of these fuels with respect to the reference fuels, *A*-3 and *C*-9. These changes are tabulated as follows:

		Road Ratings	
Fuel	A.S.T.M. Rating	Normal Heating	Added Heat
2.06 cc. tetramethyl lead	71	74	72
3.00 cc. tetraethyl lead	76	72	76

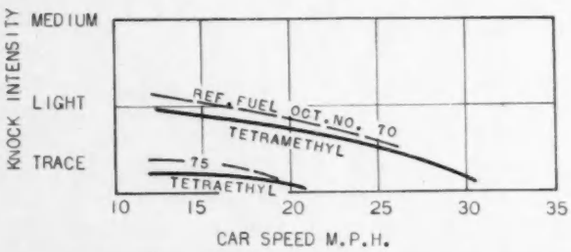


Fig. 7 – Comparison of Tetramethyl Lead and Tetraethyl Lead in a Straight-Run Gasoline – Mixture Temperature, 150-155 Deg. Fahr. at 20 M.P.H.; Other Test Conditions the Same as in Fig. 6

Thus, with normal mixture heating for this car there was an "appreciation" of the fuel containing tetramethyl lead and a "depreciation" of the fuel containing tetraethyl lead. Both of these effects were eliminated by additional mixture heating.

In actual service it is conceivable that these effects will be operative in varying degrees according to the adjustment of the mixture heat controls and atmospheric temperatures. Furthermore, the effects may be totally obscured or offset by some of the other variables mentioned in the earlier part of this paper.

These effects of vaporization on knock, both with respect to gasolines containing tetraethyl lead and to gasolines containing benzol, have a direct bearing upon knock ratings when fuels containing either of these substances are used as reference fuels in road tests. When conditions are favorable for these vaporization effects to affect knock, naturally ratings with respect to such reference fuels will be affected.

Discussion

Effect of Volatility of Antiknock Compounds Studied

– **Dr. Graham Edgar**

Director of Research, Ethyl Gasoline Corp.

THE tendency of fuels to knock is affected by a considerable number of variables even in a single-cylinder engine where the problem of distribution does not apply. When we add to these variables the distribution problem and recognize that the fuel that is actually knocking during acceleration may not have the same composition as that leaving the carburetor, the complexity of the problem increases tremendously.

The actual and relative effectiveness of antiknock compounds in suppressing knock are also affected by the variables that affect the behavior of untreated fuels. Undoubtedly, where conditions are such that the manifold does not distribute the fuel charge evenly to the cylinders, the volatility of the antiknock compound plays a part in determining its effectiveness.

The laboratories of the Ethyl Gasoline Corp. have studied this problem intensively for some time and expect, within the next few months, to present a paper giving extensive data on the subject. It is believed that this paper will represent a substantial contribution to the problem of knock suppression and will interest the refiner as well as the automotive engineer.

Additional Service Data on Effect of Ignition Timing

– **R. J. Greenshields and L. E. Hebl**

Shell Petroleum Corp.

FOLLOWING a similar line of study but on a small-sized 1936 model car, we confirmed some of the conclusions reached by the authors with larger-sized 1934 model cars, for example the conclusion regarding the effect of speed. However, our method of interpreting the results is sufficiently different that it is believed to be of interest.

As shown in the paper in Fig. 1, the ignition timing in the L-head car at 20 m.p.h. was about 11 deg. later than that giving maximum power, and it advanced at such rate that at 45 m.p.h. it equalled that for maximum power and, at 50 m.p.h., the automatic-advance mechanism had reached the end of its travel. The car we used differed from this timing as shown below in Fig. *A*.

At 14 m.p.h. the timing in our car was about 12 deg. later than maximum power, which condition agrees well with the 11 deg. for the previously mentioned L-head car. However, the rate of advance was 48 per cent lower, so the ignition continued to advance to 67 m.p.h. at which speed the timing coincided with that required for maximum power.

The question naturally arises: Since the knock vs. speed curve for a

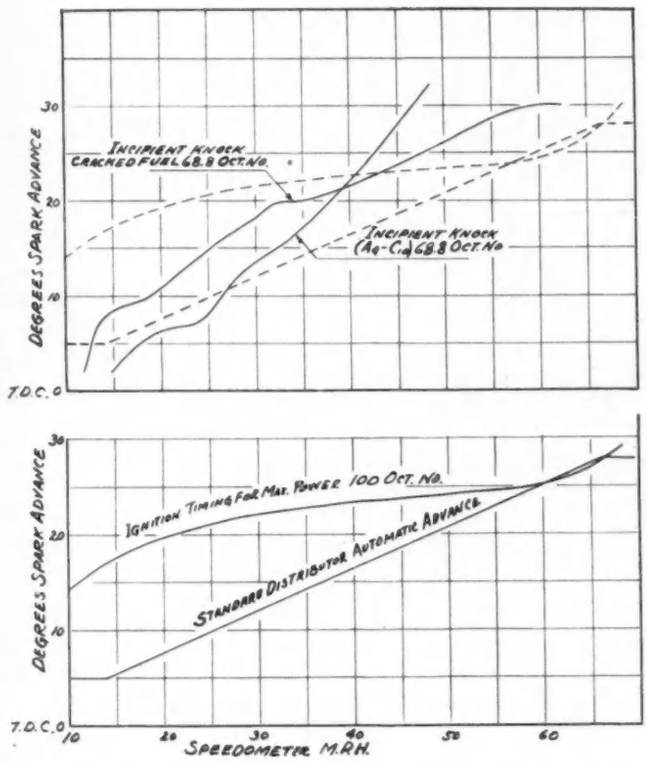

DEGREES SPARK ADVANCE
INCIPIENT KNOCK CRACKED FUEL 68.8 OCT. NO.
INCIPIENT KNOCK (A9-C4) 68.8 OCT. NO
T.D.C.
IGNITION TIMING FOR MAX. POWER 100 OCT. NO.
STANDARD DISTRIBUTOR AUTOMATIC ADVANCE
SPEEDOMETER M.P.H.

Fig. *A* – (Greenshields-Hebl Discussion) Various Ignition-Timing Curves for a Low-Priced 1936 Car

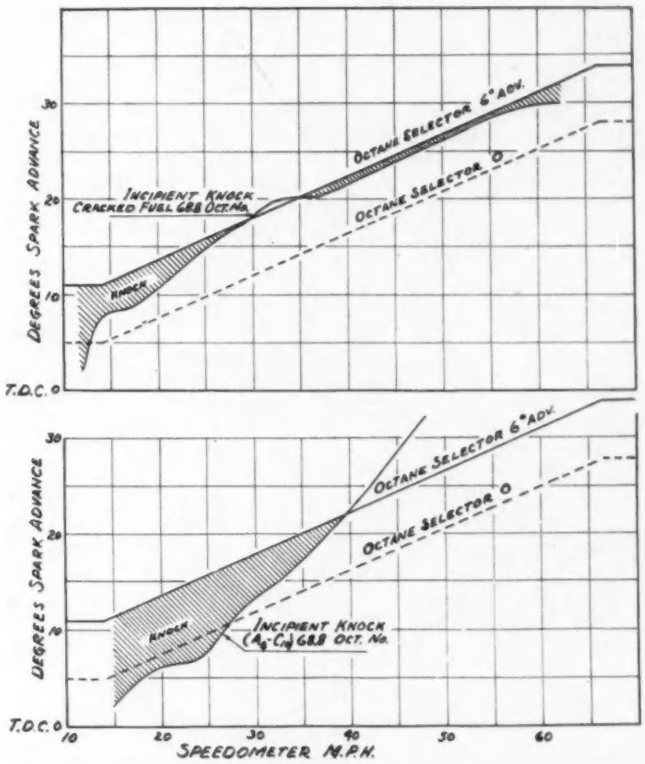


Fig. *B* – (Greenshields-Hebl Discussion) Allowable spark Advance for Incipient Knock Using a Low-Priced 1936 Car

fuel is influenced so markedly by the spark-advance factor, what is the remaining knock vs. speed relation as determined by carburetion, temperature, and volumetric efficiency if the influence of the spark advance can be eliminated? Our method of investigation is illustrated in the upper half of Fig. *A*. The centrifugal governor weights of the distributor were tied together so the ignition timing could be set manually at any fixed point. With this fixed timing the car was accelerated up a long 6 per cent grade under exactly the same conditions as for knock ratings at Uniontown, and the speed noted at which knock disappeared. This run gave one point on the curve for that fuel; the timing was then advanced 1 or 2 deg. and another knock-die-out point measured. This procedure was repeated until the entire curve of speed vs. the spark timing required for incipient knock had been determined. When the ignition timing is constant as in our tests, the knock always dies out sharply so the curve for a given fuel is very definite within about 1 m.p.h., and is reproducible as shown by repeat tests giving the same curve although run six months apart. Furthermore, the curve is free from personal error because it is not necessary for the observer to make the very difficult decision quickly whether the knock is medium-minus, or light-plus, and so on. The two curves shown are typical, the reference fuel curve being much steeper than that for the cracked gasoline.

When the characteristic curve for the fuel is known, it is a simple matter to say whether or not it will knock when the automatic-advance mechanism is used. Since the curve shows the spark advance required for incipient knock at each speed, then knock will occur whenever the ignition timing is advanced more than that shown by the curve.

The shaded areas of the two curves in Fig. *B* are knock-intensity curves, the units of knock intensity being degrees of excessive spark advance over that required for incipient knock, rather than medium, heavy, and extra-heavy plus, and so on. Obviously the speed at which maximum knock occurs is varied by a change in the rate of advance since this rate controls the distances between the two curves at different speeds. The automatic-spark-advance curve is thus really the base line from which knock intensity is measured, and this is believed to be the reason why so much difficulty is encountered in direct measurements of knock intensity – because the actual advance curve obtained in practice is influenced by mechanical variables such as looseness and sticking governor weights. Thus the base line of the knock-intensity curves varies about more or less at the mercy of the particular distributor.

Fig. *C* shows the curves of speed vs. spark advance for incipient knock for a series of reference fuels. These curves show not only that knock on such fuels was a maximum at low speeds but, what is more important, the curves have peculiar shapes. The inflections in the curves are believed to be due largely to carburetor characteristics, combustion-chamber temperatures, and volumetric efficiency, all of which are known to vary with speed and all of which remain to be investigated.

In Fig. *D* the reference fuels are compared with three commercial

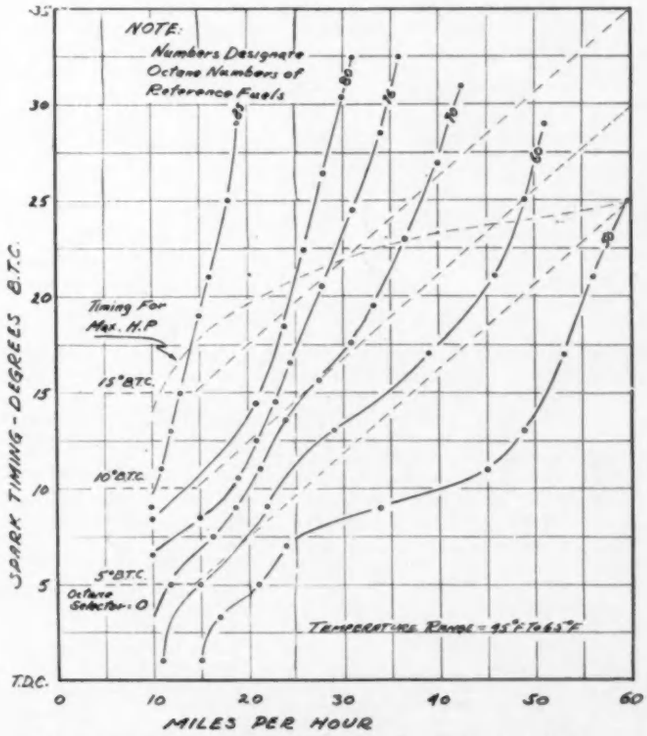


Fig. *C* – (Greenshields-Hebl Discussion) Spark Advance for Incipient Knock on *A*4 and *C*10 Reference Fuels Using a Low-Priced 1936 Car

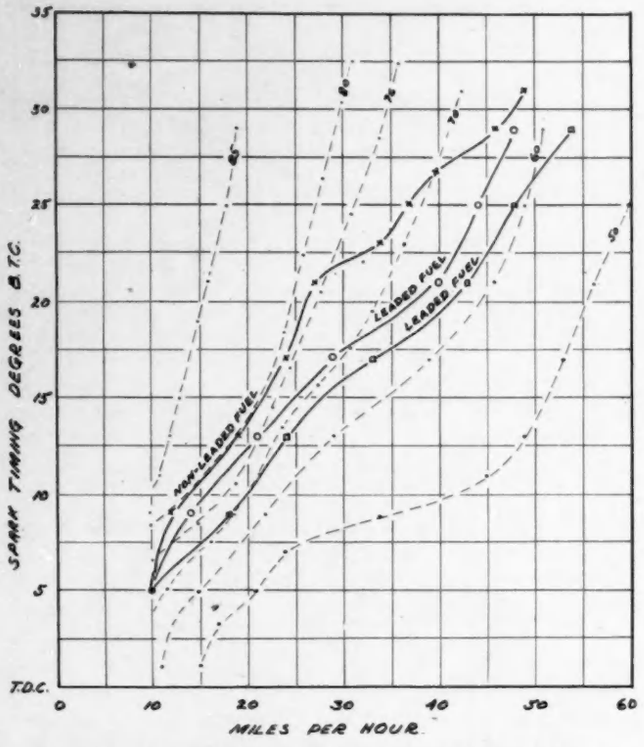


Fig. *D* – (Greenshields-Hebl Discussion) Spark Advance for Incipient Knock on Commercial Fuels Using a Low-Priced 1936 Car

regular gasolines, and the antiknock values of these fuels are seen to vary from about 73 octane number at 10 m.p.h., to values between about 60 and 65 octane number at 50 m.p.h. This result is exactly in line with the conclusions of the authors regarding the effect of speed on ratings based on direct measurements of knock intensity. The relative antiknock values indicated by our curves are strictly independent of any peculiarities in the spark-advance mechanism and, thus, the conclusions drawn previously are not subject to modification every time the distributor weights stick. It should be noted further that the curves on the commercial fuels also contained inflections that are caused by factors other than spark timing. Further study remains to be done to isolate these factors.

Influence of Combustion-Pressure Diagram of Fuel Stressed

– R. Stansfield

Chief Research Engineer, Sunbury Laboratory, Anglo-Iranian Oil Co., Ltd.

THE explanation of differences in knocking characteristics suggested by the authors is probably of major importance and a valuable contribution to our knowledge of the subject.

It seems possible, however, that there may be other factors that contribute to knocking characteristics, among these factors being the nature of the combustion-pressure diagram for each fuel type.

The difference between two types of fuel was mentioned by E. S. L. Beale and the writer in a description of a cathode-ray engine indicator, published at the end of 1935[a] and later in a more detailed way for other fuels by E. M. Dodds at the S.A.E. Summer Meeting in 1936.[b]

Fig. *E* is a tracing of a pressure diagram from the C.F.R. engine under smooth-running conditions, and below it is a diagram of rate-of-change of pressure, positive rates of change being plotted below the base line. Fig. *F* shows the diagram from the same engine, running at the same speed, compression ratio, and so on, on a knocking fuel. It will be noted that the pressure now rises much more abruptly and that the rate-of-change of pressure diagram shows not only a vibratory pressure wave following the detonation, but also an increase in rate of pressure rise immediately prior to the actual detonation.

It is found on analysis of 20 or 30 such diagrams to obtain a good average cycle that benzene blends give a rapid predetonation rate of pressure rise and a small vibratory effect following the knock onset, whereas octane blends have relatively low predetonation-pressure-rise rates and rapid combustion of the knocking portion of the fuel with high amplitude of pressure vibration. Leaded fuels that knock most heavily at slightly weak mixture strengths give low predetonation-pressure-rise rates and intermediate vibratory effects.

Examination of the knock sounds of an engine running on various fuels suggests that both the predetonation and the detonation portions of the cycle need to be taken into account. In the case of a C.F.R. engine the predetonation noise is of a relatively low frequency and gives the impression that the bouncing pin is very much underrating certain fuel types. The same fuels tested in a multicylinder engine will, however, give a much higher predetonation note, less easily distinguishable from the detonation knock itself, and just as much objected to by the car user.

Leaded fuels and those of the *C*10 reference-fuel type tend to give maximum knock when slightly weak and thus, in an average car, they usually show maximum knock just after acceleration starts when stable conditions have not been reached in the manifold. The blends rich in aromatics knock the most when the mixture is on the strong side, and this condition is attained more usually when the speed of the engine has increased and acceleration is somewhat less rapid.

The benzol fuel types, being temperature-sensitive, are also likely to develop high-speed knock more easily, and this knocking may be due in a large measure to very rapid predetonation pressure rise rather than to normal detonation.

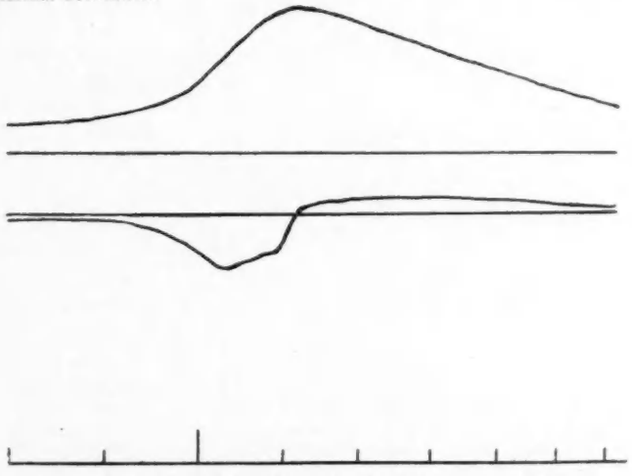


Fig. *E* – (Stansfield Discussion) Pressure Diagram (*above*) and Rate-of-Change of Pressure Diagram (*below*) under Smooth-Running Conditions – C.F.R. Engine, 87-Octane Fuel, 5.27:1 Compression Ratio, 300 Deg. Fahr. Mixture

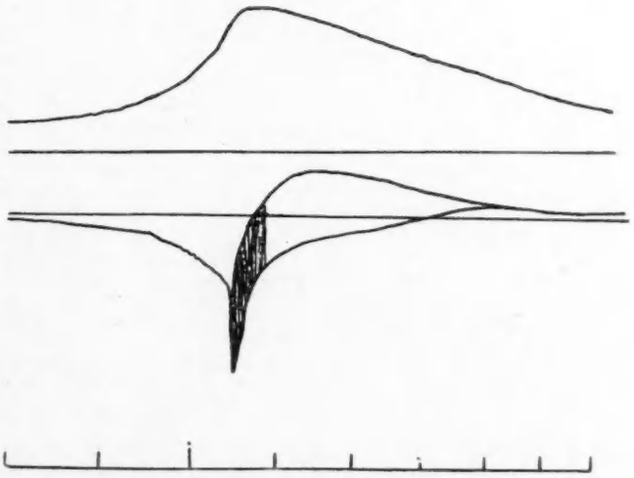


Fig. *F* – (Stansfield Discussion) Pressure Diagram (*above*) and Rate-of-Change of Pressure Diagram (*below*) Running on a Knocking Fuel (65-Octane) – Other Conditions the Same as Those of Fig. *E*

[a] See *Engineer*, Dec. 13, 1935, pp. 617-619, Dec. 20, 1935, pp. 642-644, Dec. 27, 1935, pp. 667-669; "The Standard-Sunbury Engine Indicator," by E. S. L. Beale and R. Stansfield.

[b] See S.A.E. TRANSACTIONS, December, 1936, pp. 487-495; "Development and Application of the Cathode-Ray Engine Indicator," by E. M. Dodds.

A Spark-Plug Adapted for Measuring Cylinder-Head Temperatures

By G. D. Boerlage and A. G. Cattaneo

Royal Dutch Shell Engine-Research Station

A NEW method for adapting a spark-plug to the measurement of cylinder-head temperatures is described, consisting in leading thermocouple wires through two holes down through the shell of the spark-plug and bringing them into contact with each other in the surface of the rim exposed to the combustion-chamber.

Engine tests were conducted to compare results obtained by this method of measurement with those obtained with a central-electrode couple, and also a couple incorporated in the spark-plug gasket, all three devices being incorporated in the same plug.

The thermocouple mounted as described in the paper was more responsive to engine detonation than the thermocouple mounted in the gasket but less so than the thermocouple mounted in the central electrode. It was less responsive to changes in wind speed than the gasket couple in the case of an air-cooled engine but more so than the central-electrode couple, and it is believed that a couple mounted as described gives a truer indication of cylinder-wall operating temperatures than can be obtained by either of the other methods.

IN carrying out tests on internal-combustion engines, the knowledge of cylinder-head temperatures is of primary importance. However, in water-cooled engines that are not built specially for test purposes, this measurement is difficult or even impossible as the wall of the combustion-chamber is accessible through the cooling jacket only. The inserting of a thermocouple, if possible, requires some elaborate work and involves complicated repairs in case of a failure.

These difficulties led to temperature measurements via the spark-plug.

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 11, 1937.]

FIG. 1.

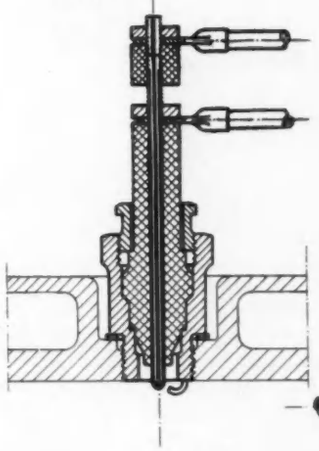


FIG. 2.

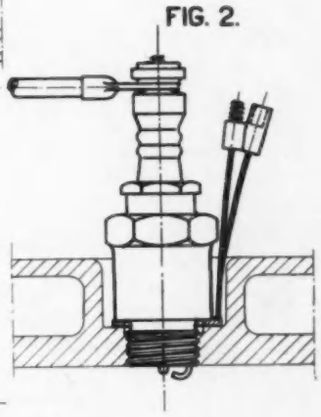


FIG. 3.

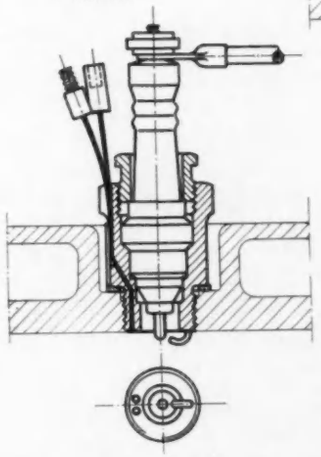


Fig. 1 – Central-Electrode Thermocouple Arrangement

Fig. 2 – Arrangement with Thermocouple Soldered to the Spark-Plug Washer

Fig. 3 – Arrangement with Couple in Face of Plug

A well-known method uses an axially perforated central electrode as one pole of a thermocouple, the second pole being an insulated wire led through the hole (Fig. 1). This system shows variations of the mean gas temperature without an appreciable time lag, but it tends to exaggerate the danger of a momentary high gas temperature. As a matter of fact, the temperature of the wall of the combustion-chamber is more important for the welfare of the engine than the maximum or the mean gas temperature. The system, moreover, has the following disadvantages: the central electrode has to be of sufficient diameter to accommodate the second wire so that ordinary spark-plugs generally cannot be adapted for the purpose; the material of the electrode should give a high thermo-voltage, a requirement not easy to reconcile with the characteristics required for an electrode; the measuring circuit comes under the high tension required for the ignition spark.

Another device that is in wide use in aero-engine practice consists of a thermocouple soldered to the spark-plug washer (Fig. 2). It has the advantage of low cost and easy installation, but has the following disadvantage: the position of the washer being nearer to the cooling medium than to the combustion-chamber wall, the thermocouple will respond mainly to fluctuations of the temperature of the cooling medium so that variation of the cylinder-head-wall temperature will be recorded on a distorted and reduced scale only. Therefore, a comparison of readings on different engines or at different places on the same engine, may be of doubtful value.[1]

An effort to avoid these disadvantages is represented in Fig. 3. The insulated thermocouple wires were introduced through two holes in the body of the spark-plug and brought into contact with each other in the rim pointing toward the combustion-chamber and lying flush with its surface. Thus the temperature in "the face of the plug" is measured.

The cost may be somewhat higher than that of the previously mentioned washer type, but this factor is largely com-

(*Continued on page* 172)

[1] See *The Journal of the Royal Aeronautical Society*, March 1, 1935, p. 895, Fig. 15; "Fuels for Aircraft Engines," by E. L. Bass.

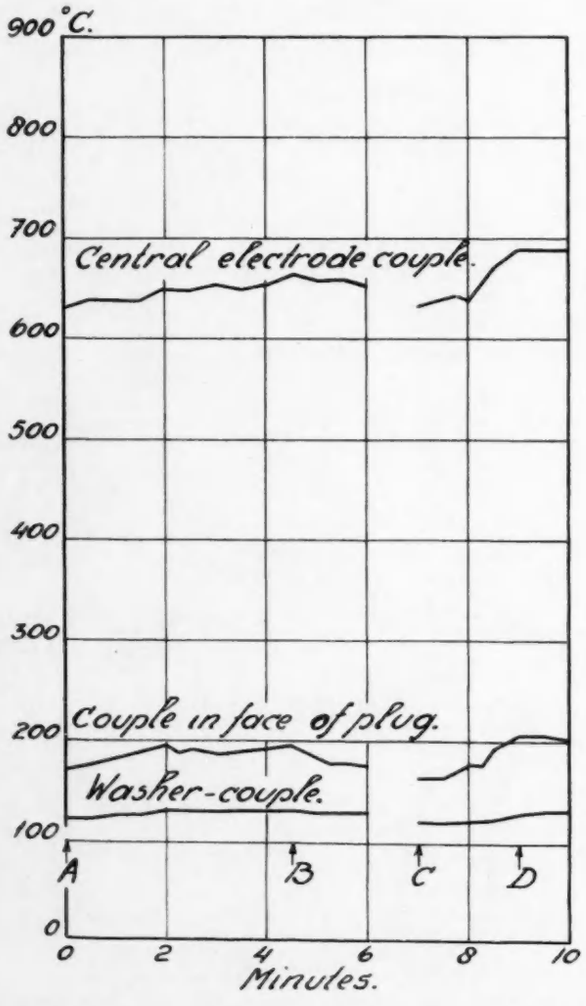


Fig. 4 – Comparison of Three Methods of Measuring Cylinder-Head Temperature Via the Spark-Plug on a Water-Cooled Engine

Ricardo variable-compression engine (E 35); piston displacement 2080 cu. cm.

A – Detonation started through low-octane fuel.

B – Detonation stopped through high-octane fuel.

C to *D* – Compression ratio gradually increased (slight to heavy detonation).

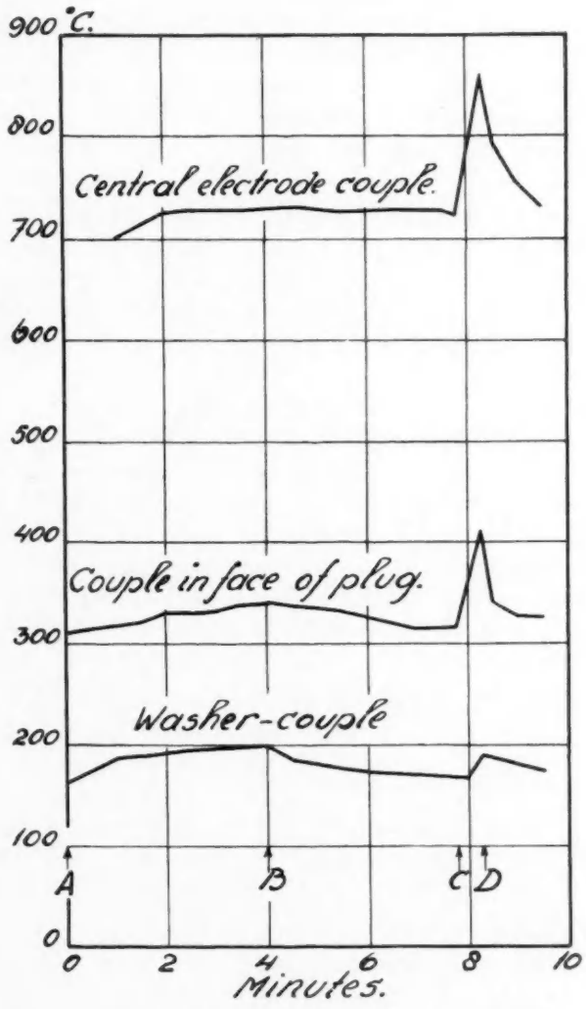


Fig. 5 – Comparison of Three Methods of Measuring Cylinder-Head Temperature Via the Spark-Plug on an Air-Cooled Engine

J.A.P. motorcycle engine; piston displacement, 500 cu. cm.

A – Wind speed decreased from 22 to 16.5 m. per sec.

B – Wind speed increased from 16.5 to 25 m. per sec.

C – Detonation started through ethyl-nitrate.

D – Detonation stopped.

Aircraft-Engine Materials

By J. B. Johnson

Chief, Material Branch, Materiel Division, U. S. Army Air Corps

THIS paper is prepared in the form of a compendium of the several materials, ferrous and non-ferrous, that are employed in the construction of aircraft engines. The data are arranged in tables giving the specifications and applications for the several metals, with descriptive matter covering the metallurgical reasoning which led to their adoption.

The limitations of the properties of the materials are stated, and suggestions made for changes in design which would permit the application of other materials. The evolution that has taken place in the development and application of metals to meet the demand of increased performance is illustrated.

THE materials used for aircraft-engine construction conform very closely to the standards published by the Society of Automotive Engineers. The allowable ranges for the several chemical elements may be restricted or changed by the engine manufacturer to meet special requirements or to maintain better uniformity in the mechanical properties of the finished product, but these changes are minor only and do not affect the basic classification of the material. This standardization is fostered by the aviation industry and by the several departments of the Government concerned, as it reduces inventories, increases the sources of supply, and would be of inestimable value in time of emergency.

The principal criterion considered by the aeronautical engineer in selecting a powerplant is the ratio of the weight to the horsepower rating. This value underlies much of the metallurgical development that has produced the modern aircraft engine. It accounts for the extensive use of alloy steels for those parts requiring resistance to wear and high specific strength and stiffness, and for the development of low-specific-gravity alloys of aluminum and magnesium with special properties for varying conditions of service.

The specifications listed in Tables 1 and 2 are those most extensively used by the manufacturers of aviation engines. The mechanical properties are typical of the specification values for several applications that have proved successful in service for engines rated from about 200 to 1000 hp. Table 3 gives the principal engine parts and representative specifications.

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 15, 1937.]

The material for any particular engine part is selected on the basis of the following factors in the order of their importance:

(1) Mechanical properties at the operating temperature.
(2) Corrosion resistance for parts in contact with fuel or the products of combustion.
(3) Uniformity; freedom from soft spots, heat-treatment cracks, and magnaflux indications.
(4) Suitability for fabrication by the methods and equipment available to the manufacturer.
(5) Machinability.
(6) Cost.

Hardness, determined by the Brinell, Rockwell, or Vickers machines, is employed more than any other mechanical test. The engineer uses it freely as a measure of the tensile strength, and the metallurgist employs it for a check of the uniformity and effectiveness of the heat-treatment. There is a tendency to consider ultimate strength and hardness as the criteria of material for engine design. It should be emphasized that there are properties in a metal that hardness tests do not indicate, such as ductility, ratio of yield strength to tensile strength, and notch sensitivity. In the case of large forgings, especially crankshafts, connecting rods, and propeller hubs, tensile and occasionally Izod impact tests are specified. Similar tests on representative specimens should be carried out on smaller parts so that there is no danger that the material is in an undesirable condition. A destructive test on a finished part will give valuable information from a study of the deflection under the applied loads and of the character of the fracture. A satisfactory test for the ductility of connecting-rod bolts, cylinder studs, and similar parts, is a simple bend test.

The Izod notched-bar impact test is specified less often than formerly for an acceptance test. It was used originally to detect temper brittleness, which may develop in the heat-treatment of the higher chrome-nickel steels. The discovery that rapid cooling from the tempering temperature would prevent this brittleness and that changes in chemical composition lessen the susceptibility to brittleness has influenced the attitude toward this test. The notched-bar test is a valuable aid in laboratory investigations of the properties of material and is one of the means of determining notch sensitivity. Steels with a low Izod value are more likely to fail under suddenly applied loads especially if there is a small defect on the surface. This tendency has been experienced with oil-hardened gears and piston-pins of S.A.E. 3250 steel and with springs and gears of 6150 steel tempered below 400

Table 1 – Ferrous Metals Used in Engine Construction

CHEMICAL COMPOSITION										MECHANICAL PROPERTIES.				APPLICATION.
CARBON	MANGANESE	SILICON	NICKEL	CHROMIUM	TUNGSTEN	MOLYBDENUM	VANADIUM	OTHER ELEMENTS	S.A.E. NO. or TRADE DESIGNATION	TENSILE STRENGTH, Lb. per Sq. in.	YIELD STRENGTH, Lb. per Sq. in.	ELONGATION in 2 IN. PER CENT	REDUCTION IN AREA, PER CENT	
0.05	0.45								1010	38,000	20,000	35		Ferrules, clips, lock wire.
0.15	0.45								1015	45,000	25,000	22		Camshaft, washers, ball ends.
0.20	0.45								1020	55,000	36,000	22		Nuts, screws, counterweights, flanges.
0.35	0.65								1035	80,000	50,000	20		Shafts, sleeves, nuts, rivets.
0.50	0.65								1050	100,000	75,000	16		Cylinder barrels, keys.
0.90	0.35								1090	225,000 to	350,000			Springs.
0.95	0.35								1095					Shims, wearing parts, valve mechanism.
0.20	0.75							S-0.11	1120	55,000				Screws, nuts, dowels for minor attachments.
0.30	0.65		3.50						2330	125,000	100,000	15	50	Bolts, studs, nuts, shafts.
0.40	0.65		3.50						2340	135,000	110,000	15	50	Connecting rods, gears.
0.15	0.45		1.25	0.60					3115	130,000	90,000	16	40	Gears, piston pins.
0.40	0.75		1.25	0.80					X-3140	130,000	100,000	17	50	Bolts, studs, shafts.
0.40	0.45		1.75	1.10					5240	135,000	110,000	15	50	Crankshaft, drive shafts.
0.50	0.45		1.75	1.10					3250	225,000	200,000	10	40	Gears, pins.
0.15	0.45		5.00						2515	170,000	145,000	14	45	Gears, cams, crankshaft.
0.12	0.45		3.50	1.50					3312	160,000	135,000	15	50	Gears, drive shafts, cams.
0.30	0.50			0.95		0.2			X-4130	95,000	75,000	12		Washers, shims, spacers, tubes.
0.40	0.65		1.75	0.70		0.35			4340	160,000	140,000	15	50	Crankshaft, connecting rods.
1.0	0.30			1.35					52100					Balls, bearings, knuckle pins.
0.35	0.75			0.95			0.2		6135	150,000	120,000	15	50	Gears, shafts, propeller hubs.
0.50	0.75			0.95			0.2		6150	220,000	200,000	10		Piston pins, gears, drive shafts, tappets, springs.
0.90	0.35			0.90			0.2		6190					Ball ends, dowels, tappets, bolts, studs.
0.43	0.55			1.6		0.4		Al 1.2	Nitralloy	135,000	100,000	18	55	Gears, cylinders.
0.35	0.45			1.2		0.2		Al 1.2	"	120,000	80,000	15	45	Piston pins, shafts, pump liners, bushings.
0.07(1)	0.45		7.0(1)	17(1)				Ti-0.2	30905	100,000	35,000	40		Exhaust manifolds, supercharger casing.
0.60	0.3			3.5	13				71360	55,000(1)				Valve, inlet.
0.65	0.3			3.5	16				71665	60,000(1)				" " tips.
0.45	0.40	3.0		8.0		1.5			Cr-Si-W	50,000(1)				Valves, inlet.
0.30	0.30	2.50	8.0	12.5					Cr-Ni-Si	60,000(1)				" "
0.45	0.50	1.25	14.0	14.0	2.5				Cr-Ni-W-Si	80,000(1)				Valves, inlet and exhaust, supercharger buckets.
1.20	0.50	1.2		18		0.6			Cr-Si-Mo	50,000(1)				" " "
1.2	0.5	0.5		13		0.9		Co-1.4	Co-Cr.	60,000(1)				" exhaust.
1.0	0.5	1.5	0.6	13.5	3.5	0.6			13.5 Cr.	50,000(1)				" "

(1) Tensile strength at 1200 deg. Fahr. - All other tensile properties are typical specification values.

deg. fahr. to attain a Rockwell hardness of C-53. Several failures occurred until the tempering temperature was raised to above 650 deg. fahr. which increased the Izod value approximately 100 per cent and reduced the hardness to C-46-48.

Practically all failures in service show typical fatigue fractures but, to the author's knowledge, a fatigue test never has been specified as an acceptance test for aircraft parts with the exception of the Stanton test, which was required by a foreign engine manufacturer for forgings purchased in this country. The failure is occasionally due to the lack of inherent fatigue strength which is the value determined in the laboratory on a highly polished specimen, but it is caused more often by stress raisers, such as laps, pits, and sharp fillets at the surface, or by seams, porosity, and lack of homogeneity within the material.

The damaging effect of inclusions on fatigue strength is not acknowledged by all metallurgists, and it is a fact that parts which have given satisfactory service may indicate an inclusion content that would cause rejection under normal inspection; yet sufficient evidence has been produced in the studies of failures to indicate that a low inclusion count is an additional safeguard in the selection of steel. The data plotted in Fig. 1 were obtained on rotating-beam specimens machined from aircraft forgings within the range of chemical composition and mechanical properties indicated. Each specimen was rated on the actual character of the inclusions at the point of fracture.

Steel of the best quality may produce unsatisfactory forgings. The common defects are surface laps below the allowance for machining, shatter cracks, and flow lines that do not follow the surface contour of the finished machined part. The first two defects can be detected by etching or by magnetic inspection, but the third must be corrected by changes in the die.

It is quite general practice to section and etch forgings in order to study the grain flow and to eliminate any lines perpendicular to the surface at the highly stressed sections, such as fillets. The examination should be made with the outline of the machined surface laid out on the etched section. The importance of grain flow on fatigue properties is illustrated in Fig. 2. The grain-flow specimens were machined from crankshaft forgings. The grain flow in a portion of the shaft was parallel to the axis and surface of the forging and, in other sections, it was at an angle, resulting in a corresponding grain flow in the specimens. The fatigue limit for the axial grain flow was normal, but the grain flow at 90 deg. reduced this value approximately 25 per cent.

Over 99 per cent of the steel used in the modern high-output engine is alloy steel. The alloy steels usually are made in the electric furnace, although very few specifications are restrictive in this matter and open-hearth steel is acceptable. The steel manufacturers supply the former in order to meet more consistently the requirements of the deep-etch test and inclusion content, which may later lead to rejection in the magnetic inspection. It is quite general practice to examine metallographic samples selected from billets representative of the heat. No standards have been established that are used by all manufacturers, but the presence of non-metallic inclusions of large size or in abnormal amounts is considered evidence that the steel is unsuitable for aircraft parts. The presence of inclusions and seams also is indicated by the light etching of a tapered or stepped-down section of a billet or bar. This test supplements the deep-etch test which is made on a transverse section cut from the end of the bar. No universal standards for deep-etch inspection are available, but segregation, pin-hole porosity, and seams are not acceptable unless they are removed subsequently by machining. If these defects are close to the surface, magnetic inspection on the finished article will indicate their presence.

The choice of a steel for a particular application is affected by sufficient variability in the factors previously mentioned so that several compositions are employed for practically identical service. The oil-hardened 3½ per cent nickel, chrome-nickel, chrome-molybdenum, and chrome-vanadium steels are employed interchangeably for parts such as bolts, studs, connecting rods, rocker arms, and small shafts. The hardness may

vary from Rockwell C-25 for nuts to C-55 for piston-pins. In the case of larger forgings such as crankshafts and propeller hubs, molybdenum is added for greater-depth hardening and machinability.

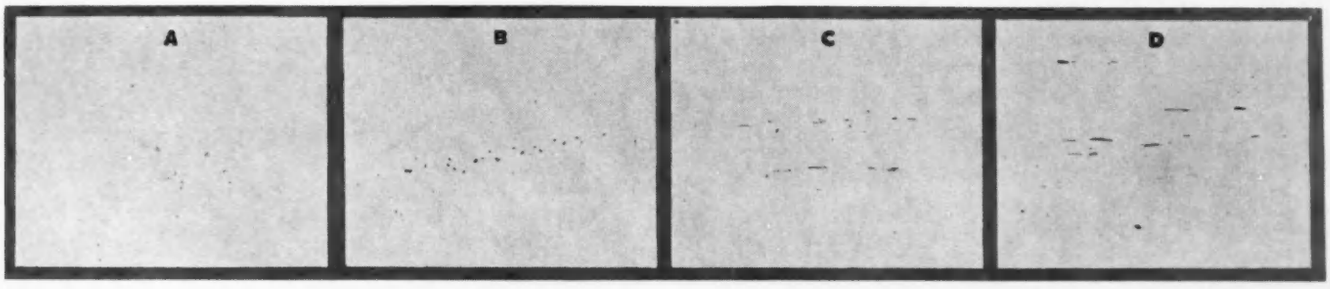
The steels selected for parts subjected to suddenly applied loads and wear, such as valve rollers, starter shafts, gears, and cams, are generally made from a case-hardened type. For the lower pressures, S.A.E. 3115, 3120 and 2315 are satisfactory, but many applications require a core strength in excess of 135,000 lb. per sq. in., which strength is obtained with an increase in the alloy content. The 5 per cent nickel, 1.5 per cent chromium-3 per cent nickel, and Nitralloy are employed extensively. For maximum core strength the upper limits of carbon content are preferred.

Small parts designed for maximum wear resistance, such as push-rod ends, valve tips, and bearings, are made from S.A.E. 71660, 6195, 52100, high-carbon tool steel 10115, and carburized 1015.

Cylinder barrels have passed through several stages of evolution both as to the method of manufacture and material. The hollow forging that is now standard practice has the advantages of uniform grain flow and minimum machining. With the lower piston speeds and side pressures of a few years past the normalized half-hard carbon steel, Brinell 180 to 200, was adequate, although somewhat better results were obtained by quenching and drawing at about 1000 deg. fahr. to a Brinell of from 225 to 260. Increase in power output required a harder barrel and chrome-molybdenum steel with a Brinell hardness of 300 came into use. The latest effort to reduce wear has brought in the nitrided sleeve with a hardness equivalent to 900 and superior corrosion resistance. Alloy cast iron is used as an alternative installation on some engines under 200 hp., but there is a weight penalty.

Valve springs are generally in locations where their length affects the overall diameter or height of the engine; consequently, the usual practice is two or three short springs nested concentrically. The maximum stress as calculated from the Rover-Wahl formula is between 50,000 and 60,000 lb. per sq. in., and the stress range from 20,000 to 37,000 lb. per sq. in., which is well within the endurance limit of small helical springs free from microscopic or macroscopic defects and operating at room temperature.

A study of valve-spring failures in service indicates that imperfections in the surface of the wire are usually responsible. This indication is corroborated by the absence of failure when precautions are taken to obtain a wire and spring free



Range of Chemical Composition and Mechanical Properties of Specimens Tested

	Tensile Strength, lb. per sq. in.	Yield Strength, lb. per sq. in.	Elongation in 2 In., per cent	Brinell Hardness Number, 3000 Kg.	Izod Impact Energy Absorbed, ft-lb.
Chrome-Nickel Steel Carbon – 0.40 – 0.46 Nickel – 1.47 – 1.96 Chromium – 0.73 – 0.84	148,000 152,000	135,000 140,000	17.0 19.0	300 325	45.0 60.0

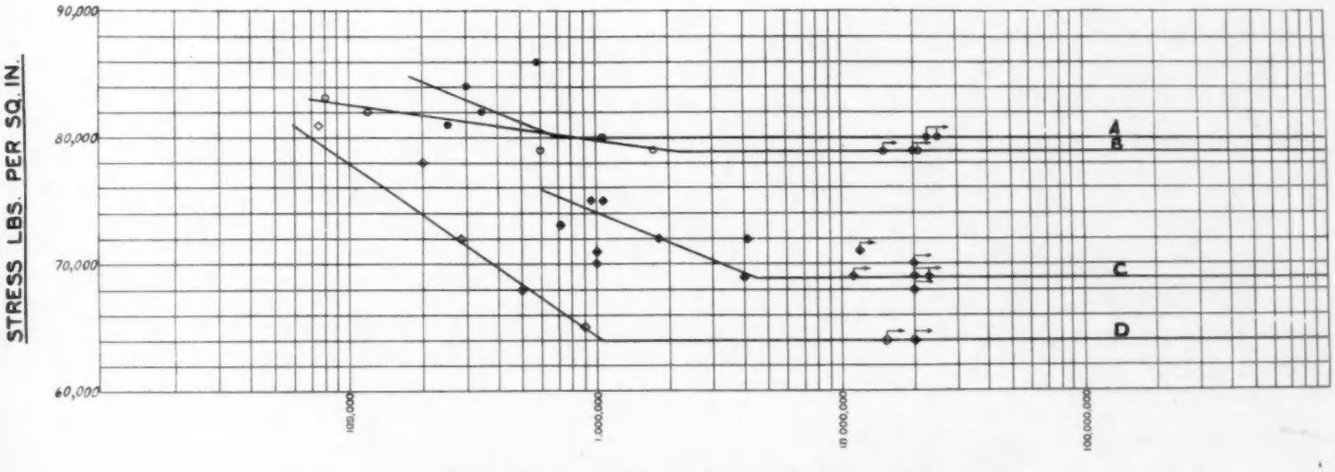
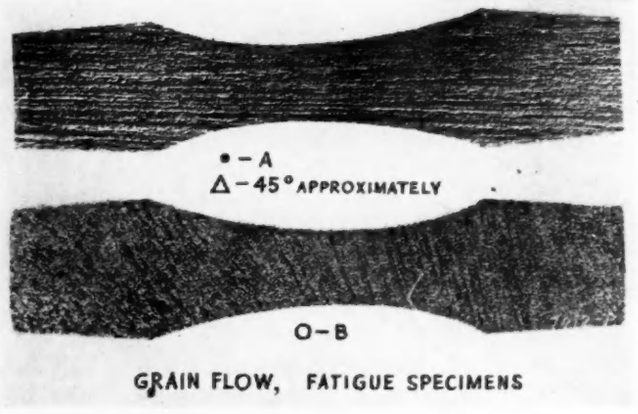


Fig. 1 – Effect of Inclusion Count on Fatigue Strength of Specimens Machined from Aircraft Forgings

from these imperfections. Special processed wire is used. The aim of processing is to eliminate decarburization, pits, scratches, and seams. Formerly, imported rods of high-carbon steel drawn to music-wire quality were employed extensively, but there has been a marked tendency to change to chrome-vanadium steel due to operating temperatures that may reach 300 deg. fahr. and to the development of a satisfactory alloy wire. The specifications require an annealed wire finished by cold drawing that will bend 180 deg. over one diameter. Samples from representative billets are examined microscopically for inclusions, and the ends of the coils are etched in 50 per cent hydrochloric acid and inspected at ten diameters for surface defects. The springs are heat-treated after coiling under practically scale-free conditions. Valve springs are inspected individually with great care. A light, wet sand blast followed by examination at 10 magnifications and magnetic inspection is applicable. The microstructures of the wire before coiling and after heat-treatment in the spring are shown in Fig. 3.

There are only a few engine parts that operate above 900 deg. fahr. These parts are the exhaust valve, exhaust stacks, parts of the turbo-supercharger and spark-plugs. The inlet valve and portions of the valve seat may reach this temperature under certain conditions. Strength, although an essential requirement, must be accompanied by non-corrodibility in the presence of gases and condensate from fuels containing relatively large quantities of tetraethyl-lead fluid. The latter has become the primary consideration in the selection of valve materials since loss in strength can be prevented by internal cooling. The modern exhaust valve for a high-performance engine, although it does its work quite satisfactorily, is a compromise from the metallurgical viewpoint. It is forged from the austenitic steel containing 14 per cent nickel, 14 per cent chromium, with or without 1 to 2 per cent silicon, which material ranks at the top for strength at elevated temperature and corrosion resistance, but lacks some of the essential characteristics of a good valve material. The seat must be faced with a hard material such as Stellite and the tip with a similar material or an oil-hardening steel in order to prevent oxidation of the former and mushrooming of the latter. The hardness of the stem is approximately Rockwell C-11 and, with any appreciable side-thrust, nitriding is desirable. This steel is subject to embrittlement at temperatures above 1000 deg. fahr. and, in thin sections, has a relatively short life. This limitation has not been a factor in valves up to the present time as sodium cooling has held down the temperature, but

Carbon	0.41	Ultimate Strength, lb. per sq. in.	139,000	Izod Impact Longitudinal Grain Flow, 59
Manganese	0.47	Yield Strength, lb. per sq. in.	115,000	
Sulphur	0.01	Prop. Limit, lb. per sq. in.	102,000	
Phosphorus	0.04	Elongation, 2 in., per cent	20	Transverse Grain Flow, 36
Nickel	1.80	Reduction in Area, per cent	55	



GRAIN FLOW, FATIGUE SPECIMENS

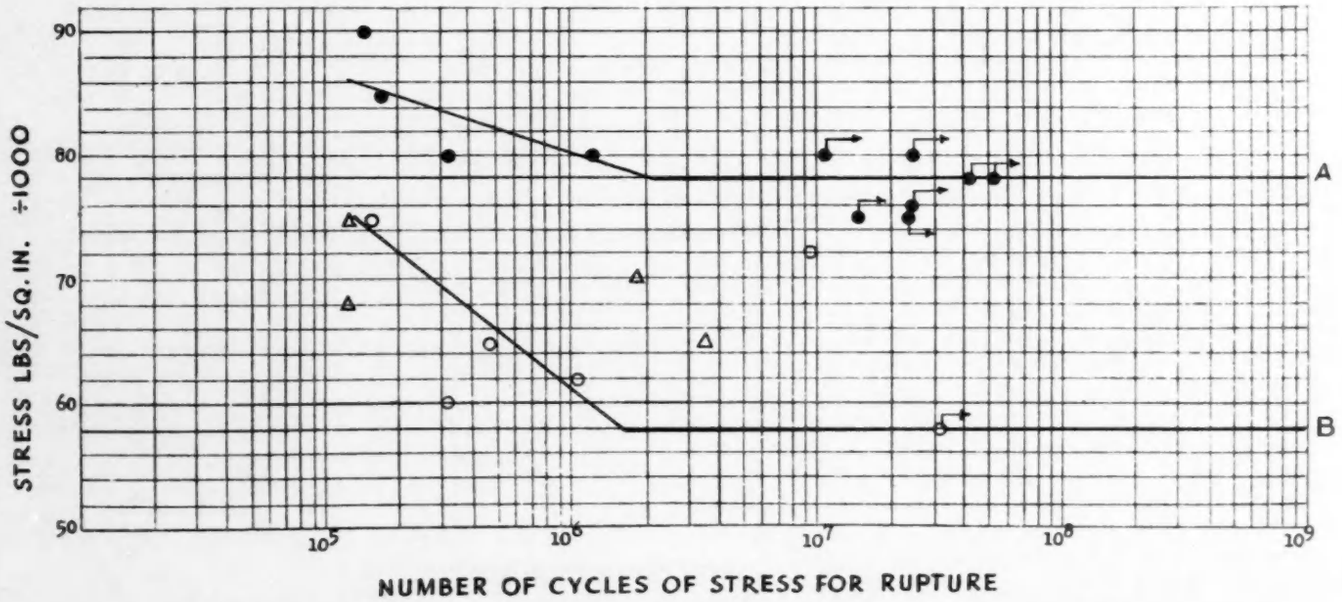
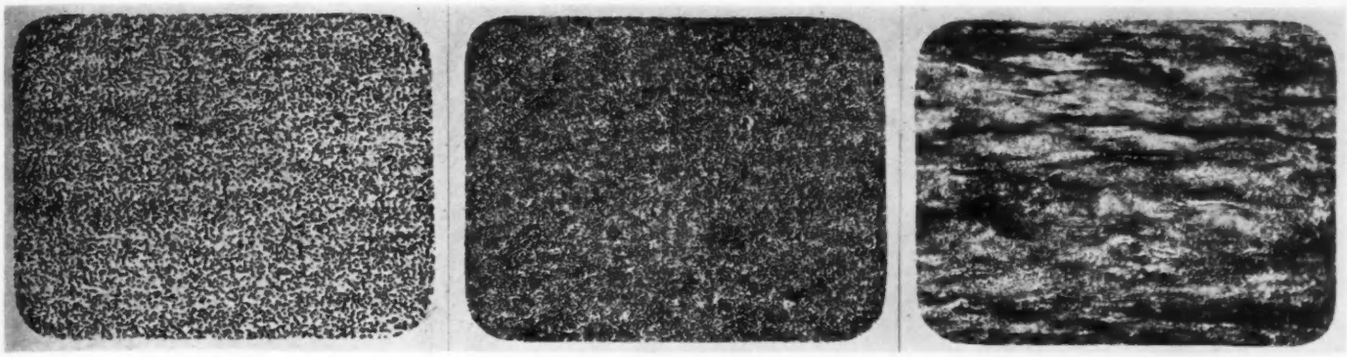


Fig. 2 – Effect of Grain Flow on Fatigue Properties of Specimens Machined from Crankshaft Forgings



	Wire, Annealed	Spring, Heat-Treated	Spring, Cold-Drawn
	Magnification 500	Etch – Nital	
Tensile Strength, lb. per sq. in.	117,000	235,000	262,000
Elongation, per cent in 2 in.	3	3	2.5
Rockwell Hardness	C-25	C-44	C-45
Carbon	0.50	...	0.92
Manganese	0.60	...	0.43
Chromium	0.95	...	...
Vanadium	0.18	...	...

Fig. 3 – Microstructures of Spring Wire – Annealed, Heat-Treated, and Cold Drawn

operating temperatures may increase unless valve stems are made excessively large.

The oil-hardening cobalt-chromium and 18 per cent chromium-molybdenum steels are satisfactory when undoped fuels are used, but will rust badly during idle periods in the presence of the tetraethyl-lead condensate. They are notch-sensitive and sharp corners should be avoided. The chrome-silicon-tungsten and chrome-nickel-silicon steels have a tendency to stretch at temperatures approaching 1200 deg. fahr. This difficulty has been experienced with inlet valves of the former material and sodium-cooled exhaust valves of the latter grade in which the head between the seat and stem was softened.

Valve steels contain more seams than the low-alloy steel, and several etchings of the valve are made before final polishing. Generally, both 50:50 hydrochloric acid and 5 per cent alcoholic nitric are used with the examination at 5 to 10 magnifications. The non-magnetic character of the austenitic steel precludes magnetic inspection.

Exhaust stacks and rings often are made from low-carbon steel, but a heat- and corrosion-resistant material such as the stabilized (with titanium, columbium, or molybdenum) 18 per cent chromium, 8 per cent nickel steel and Inconel, an alloy of nickel-chromium-iron, will give longer life and reduce the attack from atmospheric agents and from the condensate on the inside of the stack. The Inconel discolors less from atmospheric corrosion.

The weight of the aircraft engine is about equally divided between the high-density and low-density metals. In the latter category are the aluminum-base alloys for parts subjected to high fluctuating stresses and elevated temperatures and the magnesium alloys for intermediate and low stresses. The suitability of the alloy is judged by its strength at the operating temperature. This temperature for any set of conditions will depend upon the chemical composition of the alloy since it is a function of thermal conductivity and specific heat. A magnesium-alloy piston with a conductivity of approximately 0.16 cal. per sec. per cu. cm. per deg. cent., will operate at a higher temperature than the aluminum alloy with a conductivity of 0.32. The wrought alloys lose strength at a fairly uniform rate up to approximately 500 deg. fahr., whereas, for the cast alloys the rate is much lower up to 400 deg. fahr., as shown in Fig. 4. These results are from completely stabilized metal, that is, the test bars or parts from which the specimens were machined were heated at the temperature at which they were subsequently tested for a sufficient length of time to arrive at equilibrium. Only a few minor parts are used in the "as-cast" condition. Castings and forgings are heat-treated by quenching from a temperature just below the melting point of the constituents of the alloy and subsequent aging at 250 to 350 deg. fahr., although some parts simply are quenched. Attempts to employ alloys without this solution treatment for such parts as crankcases, impeller housings, and cylinder-heads have not been successful. Stabilizing treatments to reduce dimensional changes in the casting are sometimes necessary but are carried out at temperatures above 400 deg. fahr. and reduce the mechanical properties. Stabilization is done after the solution and aging treatments, and the final

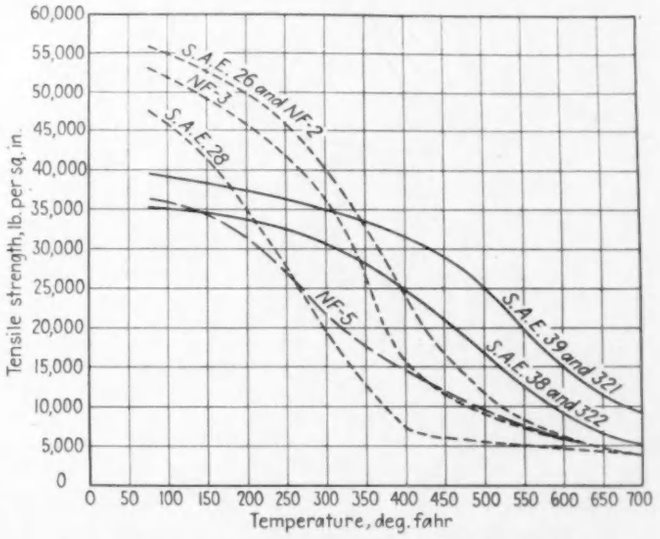


Fig. 4 – Ultimate Tensile Strength of Heat-Treated Alloys at Elevated Temperatures

Table 2 – Non-Ferrous Metals Used in Engine Construction

	CHEMICAL COMPOSITION.																MECHANICAL PROPERTIES.					APPLICATION.
	ALUMINUM	COPPER	ZINC	TIN	LEAD	SILICON	IRON	NICKEL	MAGNESIUM	MANGANESE	TITANIUM	CHROMIUM	ANTIMONY	SILVER	S.A.E. NO. OF TRADE DESIGNATION	TENSILE STRENGTH Lb. per Sq. In.	YIELD STRENGTH Lb. per Sq. In.	ELONGATION IN 2 In. PER CENT	HARDNESS(1)	FATIGUE LIMIT P.S.I. 500 x 10⁶ Cycles Reversed Bending.	
Aluminum Base	94	4							0.5	0.5					26	55,000	32,000	16	90	13,000	Washers, rivets, deflectors.
Bar	95	4.2							1.5	0.6					24	62,000	40,000	15	100	14,000	" " " small forgings.
Sheet	97								2.5			0.25			NF-1	29,000	14,000	20	60	14,000	Tubes.
Tube	91	4.0						2.0	0.6						NF-2	55,000	35,000	8	95		Pistons.
Forgings	96.5					1.0			0.7			0.25			28	43,000	34,000	12	90	11,000	Large forgings.
	85	1.0				12.0		1.0	1.0						NF-3	55,000	40,000	5	100	14,000	Pistons.
Sand Castings.	92	8													30	18,000					Cases, cover plates, oil sump, brackets.
	88	10.0							0.2						34	30,000	18,000		100		Cylinder heads.
Wgt.lb/cu.in. .092 to .105	93	4						2.0	1.5						39	32,000	18,000	1	95	8,000	Cylinder heads, pistons, bearings.
	92	4.5				1.2				0.3					38	32,000	18,000	3	80	9,000	Crankcases, gear housing.
	84	1.0				14.0		2.0	1.0						321	32,000	18,000		100		Pistons.
	92	1.2				5.0			0.5						322	32,000	20,000	2	90	7,500	Misc.castings.
	94					5.0									35	16,000	7,000	2	40	5,000	Oil pans, gear case covers.
	92					1.5			3.8						320	18,000	10,000				Castings requiring high corrosion resistance
	88								10.0						324	40,000	22,000	11	85	7,500	High stressed castings.
Die Castings.	87	4.0				5.0									307	32,000		2			Accessory housings, covers.
	87					12									305	29,000		3			" " "
Magnesium Base Forgings	6.5	0.5								0.2					NF-4	42,000	24,000	5		15,000	Nose section, forging.
Castings	10								88	0.1					NF-5	30,000	18,000	1	65	9,000	Nose section, gear case covers, housings, rear end crankcase.
Wgt.lb/cu.in. .064 to .066	8								90	0.2					NF-6	29,000	10,000	6	48	7,500	Crankcase.
	6		3							0.2					NF-7	32,000	18,000	4	65	9,000	Miscellaneous castings.
Copper Base		99.5			2.0										71-75	34,000	10,000	50		11,000	Tube, shims.
Wire		60	37												72-88	55,000	25,000	30	60		Spacers, dowels.
Sheet		66	34												70						Bolts, shims.
Tube		94		5.0									P 0.4		77&81	55-100,000		15.1		20,000	Bushings.
		98											Be 2.3		NF-8	160,000	130,000	5	300	30,000	Bushings, springs.
Wgt.lb/cu.in. .27 - .35	9	88					3								701	80,000	40,000	15	160	25,000	Valve guides, bushings.
	10	81					2.5	5		1					NF-9				200		Valve seats.
		58	42							1					NF-10	85,000	40,000	20			Counterweights.
		85	5	5	5										40	26,000	12,000	15	B-80		Fuel and oil line connections.
		88	2	10											62	30,000	15,000	14	B-85		Bushings.
		88	2	10	1.5										63	30,000	12,000	10			"
Castings		84		5	9										66	25,000	12,000	8			"
	10	89					1								68	80,000	50,000	4	170		Valve seats, propeller cones.
	11	79		5				5							NF-11	85,000		3.5	230		" " " " "
		80		12	5										NF-12						Oil seal rings.
		70		5	25										NF-13				45		Bushings.
		83		10	3			4							NF-14						Exhaust Valve Guides.
Tin Base		4.5		90									7		10	440(2)					Bearings.
				50	50															360°F.(3)	Solder.
Silver Base		30	25											45						1250°F(3)	Solder.
Lead Base				90										6		1550(2)				580°F(3)	Solder.
Nickel Base								97		3											Spark plug electrodes.
							10	70				15				80,000	30,000	25			Exhaust equipment.
Cobalt Base						1					Co-52	27	W-14						600		Stellite No.1 - Valve Tips.
						2					Co-65	27	W-4						450		Stellite No.6 - Valve Seat Facing.

(1) Brinell unless preceded by letter for appropriate Rockwell scale.
(2) Shearing strength at 350° Fahr. - Base Metal, Copper.
(3) Melting starts - degrees Fahr.

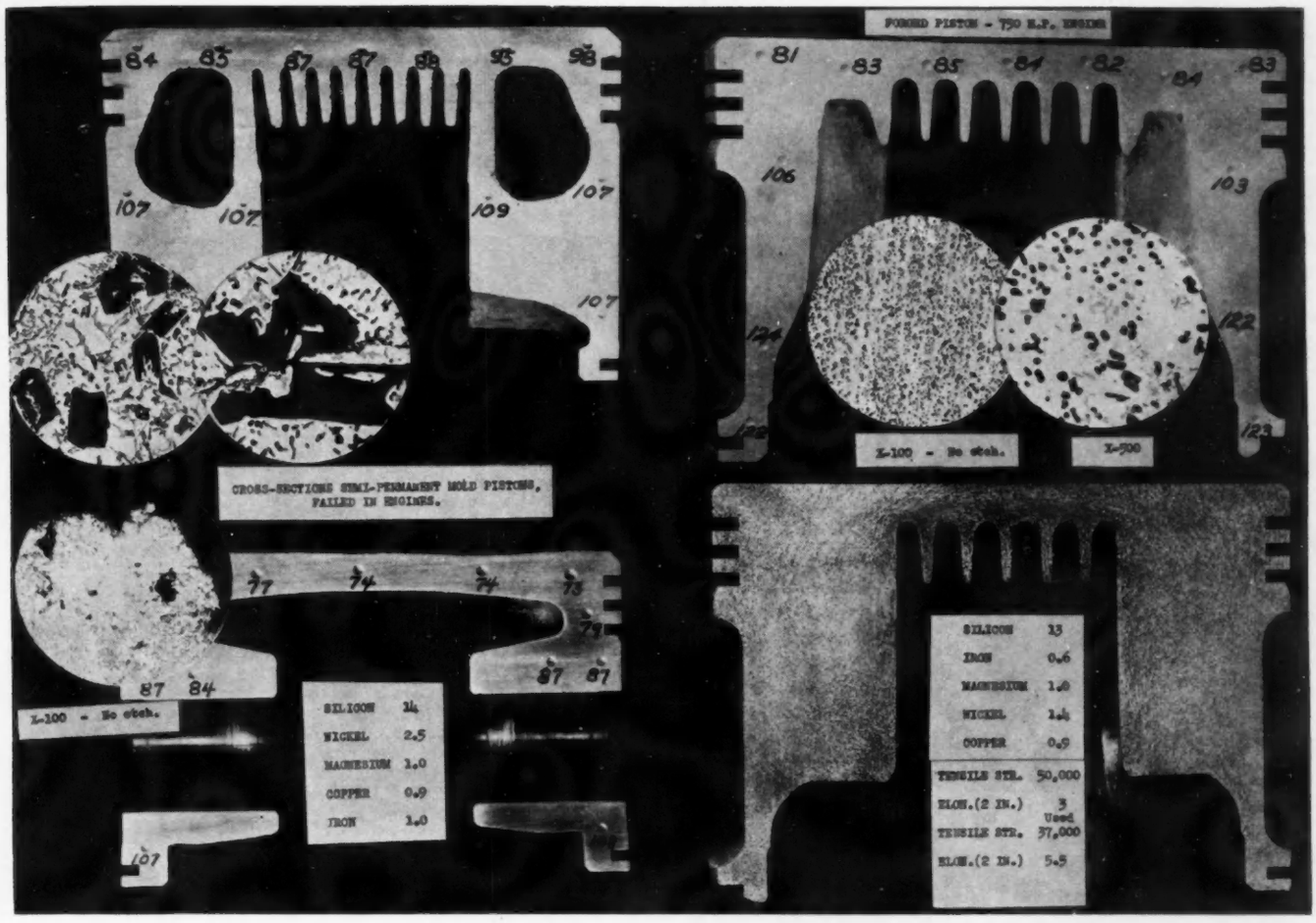


Fig. 5 – Cross-Sections, Structure, and Properties of Semi-Permanent Mold Pistons that Failed in Engines (*left*) and Forged Piston for a 750-Hp. Engine

ultimate strength may not be much greater than in the "as-cast" metal but the ratio of yield strength to tensile strength is higher.

The mechanical properties, shown in Table 2, for the cast alloys are specification values for test bars separately cast in green sand and subsequently heat-treated if required. The test-bar mold is designed to produce the maximum properties in the metal and is poured at a low temperature for the same reason. The metal in the castings seldom will have the tensile properties developed in the test bar due to changes in section, higher pouring temperatures, and variable conditions of founding. An average reduction in the specification value for tensile strength of 25 per cent may be expected in sound castings and, if porosity or sponginess is present, the reduction may be equal to 50 per cent. If the casting has large surfaces which must be maintained flat and parallel to adjoining surfaces after machining, stabilization treatments are necessary which lower the test-bar values 25 per cent and the strength of the casting a proportionate amount. It becomes difficult to estimate the strength of a casting. A laboratory test of a finished casting is a good criterion of its capacity to meet the design requirements, but it can only approximate the service loads on a complicated casting. The use of built-up determinate structures of wrought metal may offer a better solution than the large castings which become necessary with increased horsepower.

The main sections of a few crankcases for engines produced in reasonable quantities are forgings. The magnesium-silicon alloy, S.A.E. 28, modified by a small addition of chromium, has been employed on account of its forging characteristics. The tensile strength at elevated temperatures is low, and other alloys may be necessary if the operating temperatures are raised. Cast crankcases, housings, and similar parts are made from the 4 copper, 1 silicon or the 5 silicon, 1 copper alloys. The latter type has replaced the former for many applications on account of its superior founding. Castings for cylinder-heads and jackets conveying ethylene-glycol must withstand a rigorous test for leaks, which requires freedom from porosity.

Cylinder-heads are cast in dry sand from the 10 per cent copper, S.A.E. 34, and the copper-nickel-magnesium, S.A.E. 39, with either 1.5 or 0.5 per cent magnesium. The creep strengths of the alloys containing nickel are superior. The engineer is now using much ingenuity in order to prevent the temperature of cylinder-heads from exceeding 550 to 600 deg. fahr., at which point all aluminum alloys rapidly lose strength. The future of air-cooled designs may hinge on better cylinder-head materials.

Forged pistons are required for engines with high output per cylinder as the cast pistons failed in a relatively short time. The original hardness of the forged piston exceeds the cast piston but, after a few hours in an engine, the hardnesses of the head and ring grooves are comparable although the forging retains a higher tensile strength and ductility. Breaking up the coarse structure by hot work also improves the

Table 3 – Principal Engine Parts and Representative Specifications

NAME OF PART	SPEC. NO.	HARDNESS	SPEC. NO.	HARDNESS	SPEC. NO.	HARDNESS	SPEC. NO.	HARDNESS
Cylinder Barrels.	1050	225	4140	300	Nitralloy	900		
Cylinder Heads.	39	65	34	65				
Piston.	321	100	NF-2	100	NF-3	115	39	110
Piston Rings.	Cast Iron	B-100						
Valve,Intake.	Cr-Si-W	C-42	Cr-Ni-Si	C-40	Cr-Si-Mo	C-50	71360	C-45
Valve,Exhaust.	Cr-Si-Mo	C-50	Cr-Ni-W-Si	C-15	Co-Cr	C-55	13.5 Cr.	C-50
Valve,Guide.	68	175	701	160	Co-Cr	C-50	62	B-85
Valve,Spring.	1095	C-44	6150	C-44				
Valve,Spring Washer.	3135	C-35	6150	C-40				
Valve,Spring Retainer.	3250	C-50	6150	C-40				
Valve Seat.	68	225	NF-9	200	Cr-Ni-W-Si	C-15	701	160
Valve Lock Wire.	1085							
Rocker Arm.	3140	C-30	6150	C-30	2330	C-30		
Rocker Arm Hub Bolt.	3140	C-32	6150	C-25	3312	C-60		
Rocker Arm Bearing.	Ball		Roller					
Rocker Arm Cup.	10115	C-62	3250	C-42	52100	C-60	6180	C-55
Push Rod, Tube.	26		X-4130	C-35	1025			
Push Rod, Ball End.	1015	C-62	3250	C-40	1095	C-50		
Push Rod, Roller.	3115	C-60	3215	C-60				
Push Rod, Roller Pin.	3140	C-40	3250	C-50	6150	C-52		
Cam.	3250	C-55	2515	C-60				
Camshaft.	1015	C-60						
Cam Bearing.	64	B-75	62	B-85				
Cam Drive Shaft & Gear.	3312	C-60	3140	C-38	2515	C-55		
Crankcase, Main.	322	80	38	80	28	115		
Crankcase,Frt;Rear;Flower Section	322	80	38	80	NF-5 & 6	45		
Impeller.	26	115	27	100				
Impeller Shaft.	2515	C-60	3312	C-60	Nitralloy	900		
Impeller Shaft Bearing.	Ball							
Crankshaft.	X-3140	260	3240	280	4340	320	2515	C-60
Crankshaft Counterweights.	1035	160	43	140	640			
Crankshaft Extension.	3140	C-40	3250	C-45	2515	C-55		
Propeller Hub Nut.	6135	250	3312	C-60	2330	C-30		
Propeller Hub Cone.	68	210	65	160				
Rod, Connecting.	3140	270	4340	350	2340	340		
Pin, Piston.	6150	C-50	3312	C-60	3250	C-47	Nitralloy	900
Pin, Knuckle.	3312	C-60	3120	C-60	52100	C-60		
Pin, Bushings.	62	150	63	130				
Bolts, Conn. Rod, Crankcase.	3250	C-35	6150	C-35	3140	C-30		
Studs, Cylinder.	6150	C-26	3140	C-30				
Nuts.	6150	B-95	3140	C-25	2330	C-20		
Gears, Reduction; Cam Shaft Drive; Accessory Drive.	3250	C-45	3312	C-60	2515	C-60	Nitralloy	900
Housings, Accessory Drive.	30	70	322	50-80	38	50-70	NF-5 & 6	45
Housings, Accessory Drive Covers	30	70	322	50-80	35	45	38	50-70
Sump, Oil; Scoops, Air.	30	70	35	45	NF-5	45		

HARDNESS VALUES ARE TAKEN FROM SERVICE PARTS AND ARE BRINELL (10 mm.ball - 3000 Kg.load for steel and aluminum-bronze - 500 Kg. for other non-ferrous alloys) UNLESS PRECEDED BY C OR B FOR ROCKWELL - Cone and 1/16 ball, respectively.

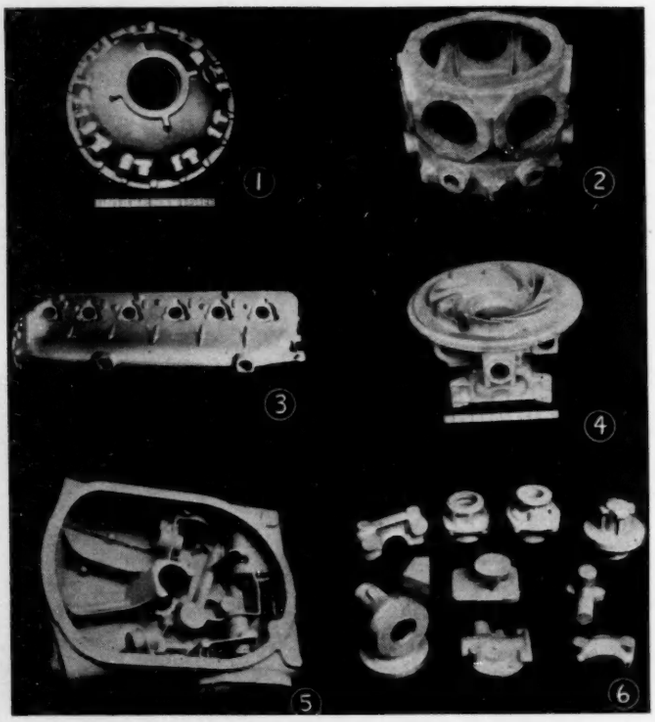


Fig. 6 – Aircraft-Engine Parts Forged or Cast from Non-Ferrous Magnesium-Base Alloys

(1) Nose Piece, Forging, NF-4.
(2) Crankcase, NF-6.
(3) Manifold, NF-5.
(4) Rear-End Crankcase, NF-6.
(5) Accessory Drive Housing, NF-5.
(6) Miscellaneous Castings, NF-5 & 7.

impact and fatigue properties and the metallographic structure, as shown in Fig. 5. S.A.E. 39 with the higher magnesium is used for cast pistons and with low magnesium it is used for forgings. The 14 per cent silicon-aluminum alloy also is applied in both forms. It has a lower coefficient of expansion but the actual difference in clearances is small, the silicon alloy taking approximately 6 per cent less clearance in the case of a 5-in. piston.

The 8 per cent copper alloys, S.A.E. 30 and 36, although still used to a limited extent on engines of low output, have been replaced largely by the 5 per cent silicon, S.A.E. 35, or the heat-treated alloys, S.A.E. 322 and 38. The silicon alloy is more satisfactory than S.A.E. 30 for thin-walled castings such as oil pans, air scoops, and similar low-stressed parts which are required to pass a pressure test for leakage. The heat-treated alloys have been substituted for brackets and housings in order to obtain sufficient ductility to prevent breakage of flanges.

Magnesium-base alloys have proved satisfactory for engine parts such as accessory-drive housings, oil sumps, and front and rear sections of radial-engine crankcases. These parts weigh less than the corresponding aluminum-alloy parts but the saving is not equal to the theoretical since the modulus of elasticity of magnesium is two-thirds that of aluminum and, consequently, larger fillets, more material in the bosses, and thicker sections in parts of the casting are necessary for equivalent stiffness. The values given in Table 2 are test-bar results, and the castings will have lower strength as in the case of aluminum alloys. Fig. 6 shows various aircraft-engine parts forged or cast from non-ferrous magnesium-base alloys.

The 8 per cent aluminum alloy with a solution treatment has high tensile strength and elongation but a relatively low

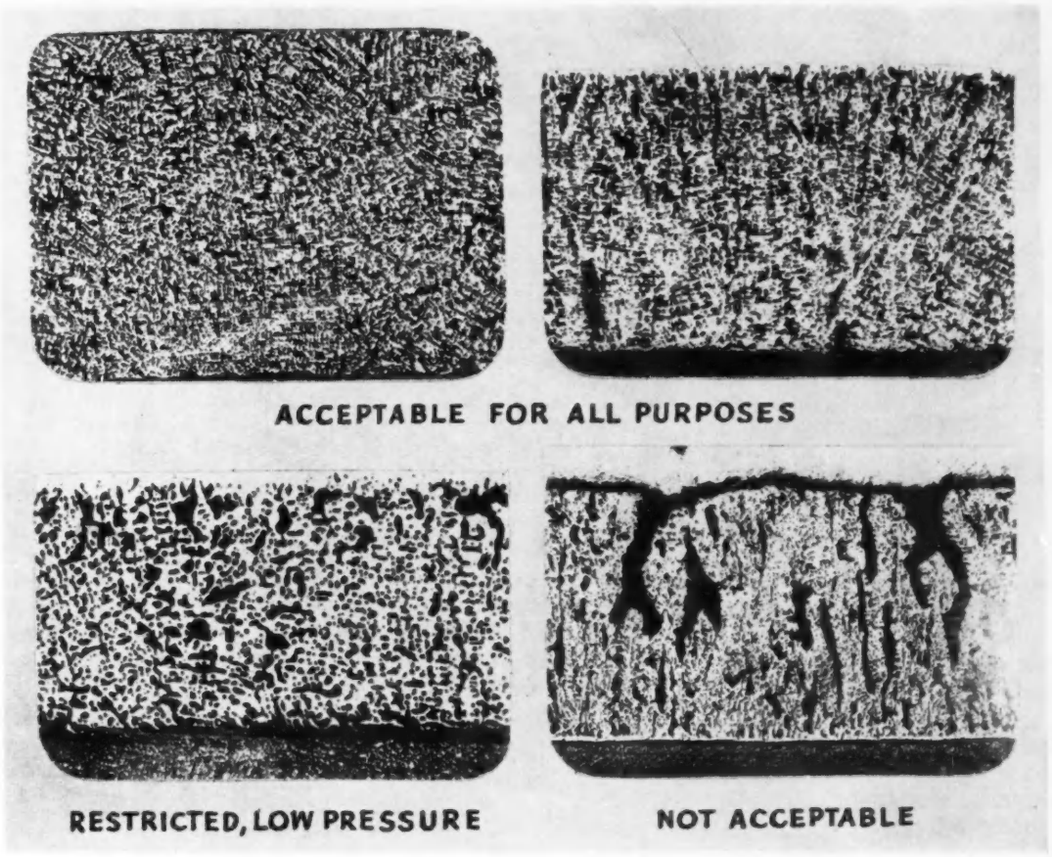


Fig. 7 – Microstructures of Copper-Lead Bronze Steel-Backed Bearings

yield strength. The 10 per cent aluminum alloy with a solution and aging treatment develops a high yield strength with lower ductility. The 6 aluminum, 3 zinc alloy solution-treated is equivalent to the 8 per cent aluminum alloy but has better resistance to salt-water corrosion.

The loss in strength at elevated temperatures and the thermal conductivity, which is about one-half that of aluminum, impair the serviceability of magnesium alloys at temperatures above 300 deg. fahr.

Copper breather lines, copper and brass shims, manganese-bronze counterweights and brass and bronze bushings and bearings are the principal parts manufactured from the copper-base metals. The physical properties of the bronzes can be varied greatly by changes in chemical composition which permit the selections of bushings for a variety of purposes. In many cases the satisfactory material is obtained by trial and error as it is difficult to correlate the conditions of service with the mechanical properties of the alloy.

S.A.E. 62 is the old standby that has given satisfactory service for bearings for pistons and knuckle pins and accessory driveshafts in the largest and smallest engines. The addition of lead up to 2.5 per cent is permissible except for the most severe conditions. Graphite-impregnated bronze is suitable for lightly loaded accessory driveshafts which are in locations difficult to lubricate.

A thin copper-lead lining, chill-cast on a steel backing, has practically displaced the tin-antimony-copper babbitts for connecting-rod crankpin bearings, plain crankshaft bearings, and many accessory driveshaft bearings which operate at pressures and temperatures at which white metal softens. The chemical composition of the bronze lining that has been used successfully in a number of engines is as follows:

Copper	69-74
Lead	26-31
Silver	1.5 Maximum
Iron	0-0.25
Nickel	0-0.1
Tin	0-0.15

The hardness and load-carrying capacity of this alloy can be raised by increasing the tin content, but scoring of the journal may occur. The penetration hardness is not a satisfactory means of selecting this type of bearing bronze. Several studies have been made of the effect of microstructure on the serviceability of the bearing, and the structures shown in Fig. 7 *(above)* have been selected as representative of bearings that are less apt to break down in service than bearings of the microstructure indicated in Fig. 7 *(below)*. The acceptable structure is dendritic with the lead broken up into fairly small particles and distributed uniformly throughout the mass. The bond between the steel and bronze should be formed by a layer of copper rather than lead. It is often possible to examine the micro-structure of the bearing without affecting its serviceability by polishing one end.

The hardness, coefficient of expansion, and corrosion resistance of aluminum-bronze led to its selection for valve seats and valve guides. Extruded bar, castings, and forgings are suitable raw materials. Guides usually are not heat-treated but installed at approximately 160 Brinell in order to facilitate line-reaming after assembly. The heat-treated bronze with a minimum hardness of 200 Brinell is preferred for seats in cylinders with high output. The composition of the bronzes that will respond to heat-treatment is quite broad. Consistent results are obtained with an aluminum content not less than 10 per cent, iron not less than 1 per cent, and nickel 0.5 to 5

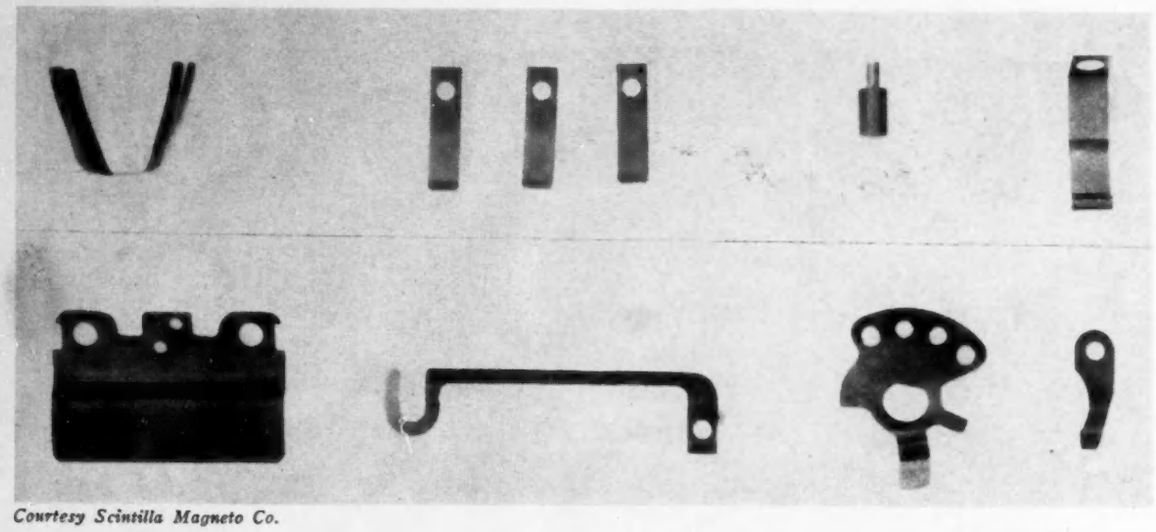


Fig. 8 – Beryllium - Bronze Springs and Clips

Courtesy Scintilla Magneto Co.

per cent. The parts should be drawn after quenching to increase ductility and to relieve quenching strains. With increased cylinder output and high spring tension, aluminum-bronzes are too soft at operating temperatures and an austenitic steel with a high coefficient of expansion similar to that used for exhaust valves, Cr-Ni-Si-W, has overcome the difficulties with the bronze. The increased stiffness in the seat material increases the life of the exhaust valve by giving more support to the head. Stellite facing has been used in some installations.

Beryllium-bronze has proved more satisfactory than phosphor-bronze for small clips and springs for magnetos and other accessories. The parts are aged after forming at approximately 525 deg. fahr. to attain a Rockwell B-110. Fig. 8 shows typical parts of this material.

Aircraft engines are exposed to the normal atmospheric variations in temperature and humidity that cause condensation of moisture with subsequent corrosion on the exterior surfaces. The condition is aggravated in the vicinity of salt water.

The most satisfactory coating for steel parts is cadmium plating. Propeller-hub parts, bolts, nuts, push rods and other exterior parts are plated with a deposit of cadmium not less than 0.0003 in. in thickness which coating will withstand a 200-hr. spray test in a 20 per cent sodium-chloride solution at 100 deg. fahr. Parts that are splashed intermittently with oil are satisfactory with a thinner deposit, and the spray test is reduced to 100 hr. Springs are not pickled but are baked for a period of 3 hr. at 400 deg. fahr. after plating to remove hydrogen embrittlement.

Zinc coatings have not been used as the thickness required is at least three times that of cadmium to give equivalent protection in salt spray.

Chromium plating does not give as good protection as does cadmium plating against corrosion from sodium chloride, but it is harder and more resistant to abrasion. It is a better coating for steel propeller blades and for re-sizing worn or undercut parts. It was tried experimentally for valve stems and piston heads, but the oxidation of the chromium at the operating temperatures of the valve and the lack of adherence to aluminum alloys made it unsatisfactory under the conditions of the test. Improvements in chromium plating may extend its field of usefulness.

Phosphoric-acid treatments are applied to air-cooled cylinder barrels to improve the adherence of the primer. The barrels are protected further by baking enamels.

Aluminum-alloy parts are anodized. The anodized layers protect the surfaces of carburetors, impellers, and similar parts from attack by water in the fuel and act as a base for paint coatings on the exterior surfaces. The chromic-acid or sulphuric-acid processes afford equal protection if a dichromate seal is used with the latter.

Magnesium-alloy parts are immersed for about 3 min. in a chromate bath, formulated as follows:

Water	8 gal.
Potassium dichromate	12.5 lb.
Nitric Acid	2 gal.

The surface forms an excellent base for primers, especially those having zinc chromate as a pigment. The zinc chromate acts as a corrosion inhibitor in seams and joints and is superior to other primers in this respect.

The protection of magnesium is not entirely adequate. A short immersion in salt water due to a forced landing will generally cause serious pitting of magnesium-base castings. An anodic process has been developed using an aqueous electrolyte of sodium bichromate and monosodium phosphate, which may prove to be more satisfactory.

Discussion

Supplementary Information on Light-Alloy Castings

– G. D. Welty

Aluminum Co. of America

SO far as the light alloy section of Mr. Johnson's paper is concerned, I can say that this sifting and selection of material and its presentation have been most skillful. In commenting on the paper I shall only attempt to offer some supplementary information bearing on a number of the points raised.

Attention frequently has been called to the discrepancy that exists between physical properties obtained from test bars individually cast and similar properties obtained from bars cut from actual castings, both poured from the same heat of metal. In the case of sound castings this discrepancy is given as about 25 per cent and, in the case of castings of questionable soundness, as much as 50 per cent. The reasons usually advanced for such discrepancies are: (1) the necessity of pouring the

casting at a higher temperature than is required for the individually cast test bars; (2) the presence of thick and thin sections in the castings which affect the rate of cooling, and (3) inability to "feed" the casting as adequately as the test bar during the solidification period.

Although all of these things do tend toward making the physical properties of the metal in the castings lower than those of the test bars, there are other factors sometimes of even greater importance that are overlooked frequently and that should receive the most careful consideration.

When test bars are cut out of a casting, it is usually for the reason that the casting has failed in service and somebody is trying to discover why it failed. Frequently the physical properties of test bars so obtained are hardly within gun shot of the required minimums written into the specifications for the individual bars, and it is assumed that the difference is due either to poor foundry practice or to the casting just naturally having very much poorer properties than those of the corresponding test bars. As a matter of fact, the major part of the difference may be due to the effects of engine operating temperatures or to processing of the casting during the course of engine manufacture. In the case of cylinder-heads, for instance, it is necessary to heat up the castings for the shrinking-in of valve-seat inserts, valve guides, cylinder barrels, and spark-plug bushings. These operations should be carried out at the lowest practical temperature and in the shortest possible time if the lowering of physical properties is to be held to a minimum.

I recall the case of a cylinder-head casting that failed in service; test bars cut from this casting showed a strength scarcely more than half of that obtained from the individually cast bars. By careful control of each step in the foundry and heat-treating practice it was possible to produce this casting so that test bars cut from any part of it showed properties in excess of 85 per cent of those of the individual test bars. In the course of manufacture, this casting was heated twice to a temperature of 575 deg. fahr. which was thought to be necessary to carry out the shrinking-in of inserts. The result was that bars tested after the inserts were put in showed a decrease in tensile strength from over 30,000 lb. per sq. in. to about 20,000 lb. per sq. in. with a proportionate falling off in yield strength and hardness. By adjustment of the shrink fits and by carrying out the entire operation in one heating at 475 deg. fahr., a strength of over 27,000 lb. per sq. in. was maintained in the casting, which value represented a relatively small sacrifice in properties.

Mr. Johnson points out that many important aircraft castings are given a solution heat-treatment and that subsequent stabilizing treatments are sometimes required to insure against dimensional changes upon machining. These stabilizing treatments, frequently requiring considerable periods of time at temperatures over 400 deg. fahr., have a considerable effect in reducing mechanical properties. I am well aware that much distortion of aluminum castings has been encountered upon machining and that many manufacturers have accepted reluctantly the lower mechanical properties that go with "stabilizing" because there seemed to be no other way out. However, much has been learned during the past few years about heat-treating and, in the great majority of cases, it is now possible to so heat-treat castings as to maintain a low level of internal strain, which will allow machining without warpage and this advantage at practically no sacrifice in the mechanical properties, thus rendering stabilizing treatments less necessary.

Cylinder-heads represent approximately one half of the light-alloy weight of the air-cooled engine. Specific gravity, thermal conductivity, and strength at elevated temperatures are the major considerations affecting the choice of cylinder-head alloys. Unfortunately some of the best alloys from the standpoint of strength at elevated temperatures are the heaviest or the poorest for thermal conductivity, which condition means that, under a given set of conditions, they will operate at so much higher temperature that the advantage of their extra strength is lost. Everything considered, Y alloy containing 4 per cent copper, 2 per cent nickel, and 1.5 per cent magnesium probably offers as good a balance of essential properties as any material yet developed for air-cooled cylinder-head applications.

The superiority of forged over cast pistons is so well established that no comment beyond that already made by Mr. Johnson seems necessary.

The use of magnesium-base alloys in aircraft-engine construction is increasing and will continue to increase. At the present time the applications are practically all in the form of castings but we may expect to see magnesium-alloy forgings and pressings enter into the field in the future. The use of the metal almost certainly will be confined to parts operating at relatively low temperatures as there seems to be little prospect that any magnesium-base alloy will be available that can compete with aluminum for the high-temperature applications such as pistons and cylinder-heads. As Mr. Johnson points out, the elastic modulus of magnesium is low – about 6½ million as compared with 10½ million for aluminum. This value means that the deflection under a given loading is greater and that the material is consequently more notch-sensitive than is aluminum. It is usually disastrous to attempt the substitution of magnesium for aluminum from the same die or pattern equipment; however, if the properties of the metal are taken into consideration in design, it frequently is possible to make an entirely satisfactory substitution for aluminum and at the same time effect a substantial saving in weight. With weight-saving of such vital importance in all aircraft construction, it seems certain that we may expect a wider application of magnesium alloys as designers acquire more experience with their properties and characteristics.

One of the great objections to the use of magnesium alloys in the past has been their susceptibility to atmospheric corrosion, particularly to moist salt atmospheres such as are encountered in the tropics. Much effort has been directed toward solving this corrosion problem, both from the standpoint of providing more satisfactory protective coatings and from the standpoint of improving the inherent resistance of the alloys themselves. Most of the magnesium-alloy castings used in the past have been made from compositions containing from 8 to 10 per cent aluminum with the remainder of magnesium. Lately, however, there has been developed the alloy containing 6 per cent aluminum-3 per cent zinc, mentioned in Mr. Johnson's paper, which alloy is unquestionably superior from the standpoint of corrosion resistance and which also possesses a much better endurance limit. This alloy, covered in specification M-112d, will be required in all magnesium castings supplied under Navy contracts after Jan. 1, 1937.

It is believed that the dichromate bath composition given in Mr. Johnson's paper is somewhat inferior to a bath made up in accordance with the following formula:

Sodium dichromate	1.5 lb.
Concentrated nitric acid	1.5 pt.
Water	to make 1 gal.

It will be noted that the principal difference between this formula and that mentioned in the paper is the lower acid concentration which seems to produce a more roughly etched surface and, hence, affords a better tooth for paint. It is common practice to sand-blast rough castings prior to treating in the dichromate bath in order to insure a uniform etching effect and a surface to which paint will readily adhere.

There is still another type of coating applied to magnesium castings which is known as the alkali-dichromate treatment, recently covered in the Bureau of Aeronautics specification M-303-2. This coating is applied by immersing the casting in a boiling solution of the following composition:

Potassium dichromate	1.5 per cent by weight.
Ammonium sulphate	3.0 per cent by weight.
Ammonium dichromate	1.5 per cent by weight.
Ammonia (specific gravity – 0.880)	0.5 per cent by volume.
Water	remainder.

The time of treatment is usually about 30 min. This treatment produces a coating fully equivalent as a base for paint to the coating applied by sodium-dichromate, nitric-acid dip and has an additional advantage in that it can be used without danger of altering machined dimensions.

Numerous tests on many types of paint have shown that, under severe conditions of exposure, top coats of an oil-base phenolic-resin enamel or of an oil-base phenolic-resin varnish pigmented with aluminum powder, render the best protection. Baking of the primer and of each finish coat is beneficial, and three top coats over the primer are recommended. Although it is true that two finish coats will, in most instances, provide satisfactory protection, the additional protection of a third coat is well worth the extra expense, particularly on parts to be used on Navy aircraft.

Contributes to Data on Steel-Backed Bronzes

– George A. Zink

Allison Engineering Co.

I BELIEVE that more data on the physical properties of the steel-backed bronzes used as bearing materials is in order.

A comparison of the Brinell hardnesses at temperature of the copper-lead and copper-lead-tin bronzes referred to by Mr. Johnson follows:

Temperature, deg. fahr.	Brinell Hardness	
	Copper-Lead	Copper-Lead-Tin
72	26.0	44.5
150	25.5	42.0
200	23.5	41.0
250	22.0	39.0
300	20.5	39.0
350	19.0	38.2
400	18.0	37.2
450	16.5	36.5
500	14.0	35.4

These readings are an average made from ten different specimens of each type of bronze at the temperatures specified. All results recorded were within ±5 per cent of the average results recorded in the foregoing

table. The hardness specimens were chill-cast on a steel back. The thickness of bronze was 0.125 in., the steel 0.250 in.

The chemical composition of the copper-lead specimens was within the limits given in Mr. Johnson's paper. The chemical composition of the copper-lead-tin specimens was within the following limits:

Copper	68.0-72.0
Lead	23.0-27.0
Tin	2.0- 4.0
Impurities	0.25 maximum

One method of comparing the relative scoring characteristics of the two bronzes consisted in cutting off the oil supply to a 2-in. bearing rotating at 3400 r.p.m. under a load of 3400 lb. per sq. in. of projected area. The length of time each bearing ran before seizure was used as an index of the seizing characteristics. The bronze with the tin addition seized in one-fourth the time required to seize a copper-lead bronze. Possibility of seizures of copper-lead-tin bearings can be reduced materially by increasing the diametral clearance. A diametral clearance of 0.0015 in. per in. of diameter is used in practically all installations of copper-lead-tin bearings.

The micro-structure shown in Mr. Johnson's Fig. 7(*a*) seems to give the best frictional characteristics to a bearing, but the resistance of such a structure to fatigue is questionable. From the performance of copper-lead-tin bearings it would seem that a spheroidal rather than a dendritic structure is desired for resistance to fatigue.

Considers Hidden Defects in Steel

—Henry Wysor
Metallurgical Engineer, Bethlehem Steel Co.

FROM the standpoint of a steel engineer, I consider Mr. Johnson's generalizations on hidden defects most important.

We may simplify our definition of a defect in a steel part as irregular or interrupted continuity of metal causing over-stress and over-strain. Obviously, the extent and locality are vital factors in the initiation of the progressive or fatigue type of rupture.

After taking care of the more obvious things, such as design and defects on the surface, we attempt to spot internal defects by means of indirect destructive tests and by non-destructive tests applied to finished parts. The difficulty lies in the interpretation of results so obtained. If the results are not quantitative, the engineer naturally is inclined to play safe, although he still tries to evaluate the data. The relative effects of inclusions with respect to size, shape, frequency, and location may be important although difficult to determine and, since inclusions cannot be eliminated entirely, something is always left to the decision of enlightened opinion. Herein lies the peril of being misled and in different directions. The inclusion itself may not be so important as the ferrite band in which it lies, and the effect of both diminish rapidly as depth from the surface increases.

Also, porosity, whether from solid inclusions or gas, may be advantageous to a degree. In many instances longer life under both static and dynamic stress conditions has been associated with a porous as compared with a dense steel. It is fortunate if the material in the actual section or in the part itself can be subjected in test to equivalent service conditions.

Case-Hardening and Chromium Plating Data Requested

—H. K. Cummings
National Bureau of Standards

I SHOULD like to ask the following questions:

(1) How extensive is the use of case-hardened crankshafts in aircraft engines?

(2) Does warping during case-hardening produce any difficulty in finish-grinding to a uniform case thickness?

(3) Has the Air Corps any information regarding the effect of chromium plating on the fatigue limits of welded hollow-steel and solid-steel propeller blades?

In connection with the third question, the Division of Metallurgy at this Bureau has found the fatigue limit of steel blades to be reduced as much as 50 per cent by thin coatings of chromium, although the reduction was considerably less with thicker coatings. The fatigue limit of welded-steel blades is usually determined by the weld, but this effect of chromium plating may be serious in the case of solid-steel blades.

Modern Body Structure and the Service Problem

TO some people rate-making for automobile insurance companies appears difficult and involved. That is not true. To establish a rate through experience is a fairly simple matter. The manual rates of the conference companies are, of course, established on the composite experience. Other companies that have sufficient business themselves and, especially those that are selecting their risks carefully, gather enough experience to establish their own rates. A rate is established practically by taking the cost of any particular coverage over a period of time sufficient for safety and adding thereto administration costs.

Consequently, it is apparent that if the cost through adverse experience of insureds having accidents increases, the rates charged for collision and property damage insurance will increase. Automotive engineers are interested in this question because, after all, the cost of purchasing and maintaining an automobile is what the customer is interested in. Unless something drastic can be done to reduce automobile accidents, there is no question that the repair bills of the American public will increase and so will their insurance rates. There is less money to spend, therefore, for something else. If the design of cars can aid in keeping down or even in preventing certain types of accident, that much money of the American public will be saved for either the purchase of new automobiles or some other worthwhile commodity.

I am convinced that the cost of making repairs from an insurance angle has increased from 15 to 25 per cent in the last three years. Now this increase in cost is not entirely due to an increase in accidents. Part of it is due to the fact that certain parts of a car are now damaged that heretofore had not been. For example, a great deal of difficulty is encountered in repairing knee-action equipment and also the grille work on radiators. These grilles are now constructed so that in parking they are damaged and are not easily repairable. Consequently, the insurance company must entirely replace the grille at an additional cost. Probably the public has demanded the particular designs that are being produced. There may be no way in which engineers can relieve the situation. If they cannot, of course, these repairs and replacements will have to be paid for by the public.

If the body is to be so constructed as to be minus what is known now as a chassis, then only experience over a proper length of time can demonstrate what the real effect will be on insurance rates. Insurance companies, however, can safely anticipate some of the difficulties that may arise.

In case an automobile having a modern unit body is entirely destroyed, there is no more problem involved than in any other type of construction as far as the insurance company is concerned. It must pay for the value of the car or replace it. The difficulty arises, however, in various types of accidents that in themselves may be classed as minor. If the modern body is to be so constructed that repairs cannot be easily made and the whole body must be junked and a replacement made, then anyone can reasonably determine that the rate on collision on this type of case would be considerably affected and the increased cost must be paid by the public.

Excerpts from the paper, "Modern Body Structure and the Service Problem," presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 13, 1937, by Howard D. Brown, general attorney, Detroit Automobile Inter-Insurance Exchange.

Engine Temperature as Affecting Lubrication and Ring-Sticking

By C. G. A. Rosen
Caterpillar Tractor Co.

IN the investigation of combustion in Diesel engines considerable emphasis was placed upon temperatures prevailing in metal parts adjacent to the combustion-chamber envelope. In mobile-type Diesel engines these temperatures are influenced directly by design characteristics, service conditions, atmospheric temperatures, and operating schedules. Of particular importance is the combined effect of the factors upon piston temperatures and, therefore, of direct consequence to lubrication.

Oxidation tests of lubricating oils for metals have demonstrated that gummy and carbonaceous products are deposited in relation to the heat gradient in the piston, particularly in the ring-belt region. The type and extent of these deposits are influenced further by the source of the crude and by the method of treatment and of finishing the lubricating-oil stock.

The gummy deposits from the lubricating oil act as binders to congeal carbon and dust in the ring-grooves to produce ring-sticking. Even though mechanical construction and thermal regulation can provide some alleviation for ring-sticking, the great variety of service, operating, and atmospheric-temperature conditions encountered in Diesel tractors operating throughout the world calls for specialized lubricants.

This paper deals with the program of investigation of the influence of engine temperature on lubrication and ring-sticking and the development of improved lubricants for high-duty Diesel service.

IT is a well-recognized axiom that the oil man is blamed for most of the ills that befall mechanical equipment. Any mechanical field failure involving the use of lubricants invariably draws first fire on the oil. Often this procedure is a preliminary barrage to cover equipment defects. Sometimes it is used as a sustained excuse for lack of knowledge concerning the reason for difficulties.

With full cognizance of this situation, the Caterpillar Tractor Co. attacked the Diesel ring-sticking problem as one requiring careful investigation of its own product, concerning the following influential factors:

Table 1 – Factors Investigated

(1) The combustion process.
(2) Design characteristics of liner, piston, and piston-ring.
(3) Metallurgical factors as affecting rubbing surfaces.
(4) Machining operations as influencing lubrication.
(5) Break-in and testing procedures, as determining surface conditioning.

According to Boerlage & Broeze the Diesel combustion process may cause the formation of three types of products of incomplete combustion upon the surfaces surrounding the combustion chamber, such as:

(1) Aldehydes and varnishes, produced by chilled hydroxylation.
(2) Soots from over-rich mixtures.
(3) Formation of products by chilled destructive combustion.

These factors are influenced considerably by the characteristics of the fuels that are burned. They are not related to the lubricants used. A fourth type of deposition may be produced by the decomposition of lubricating oils under the influence of the combustion of Diesel fuels. These decomposition products accumulate soot from the fuel and build up sticky materials on the piston-ring lands and in back of the piston-ring grooves.

The combustion process in the Caterpillar Diesel was investigated thoroughly for the purpose of determining the

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 13, 1937.]

products of incomplete combustion of fuel. When burning specification distillate fuels within the combustion-chamber temperature ranges that prevail in field operations, the formation of aldehydes could not be detected, whereas the soots produced by chilling influences at low jacket temperatures and light loads were of minor consequence in the formation of depositions found on cylinders, pistons, and ring-grooves. A number of experiments were conducted with various types of piston crowns and design structures adjacent to the combustion-chamber envelope, thus producing wide ranges of temperature of surfaces adjacent to the combustion-chamber. It was concluded from these tests that fuel-deposition products vary in their formation, character, and quantity in relation to the type of combustion system employed and to the engine factors that enter into their control. In general, the temperatures prevailing in metal parts adjacent to the combustion-chamber envelope have a direct influence upon the formation of deposits within the combustion-chamber. For subsequent tests, these temperatures were controlled by design characteristics so as to produce a minimum of deposition from products of incomplete combustion. The ring-belt temperature, however, was found to have a decided influence on the products of decomposition of the lubricating oil.

In exploring factors (2), (3), and (4) in Table 1, which factors may be said to be the structural elements involved in ring-sticking, it was found that ring-sticking had a powerful ally in the seizure of rings, namely, ring-scratching or surface abrasion of the ring face rubbing against the cylinder wall. The material that abraded or burnished off the ring and cylinder would fill the ring-groove clearance and clamp the piston-ring securely in the piston. It was, therefore, apparent that two important conditions must be met to insure long life of cylinder and rings:

In the first place, reciprocating rubbing surfaces must be transformed from raw-machined and process-finished materials to surfaces capable of maintaining an adequate and suitable oil film. This subject is one of paramount interest to the industry, and one about which so little is known definitely. In the second place, the structural factors adjacent to the

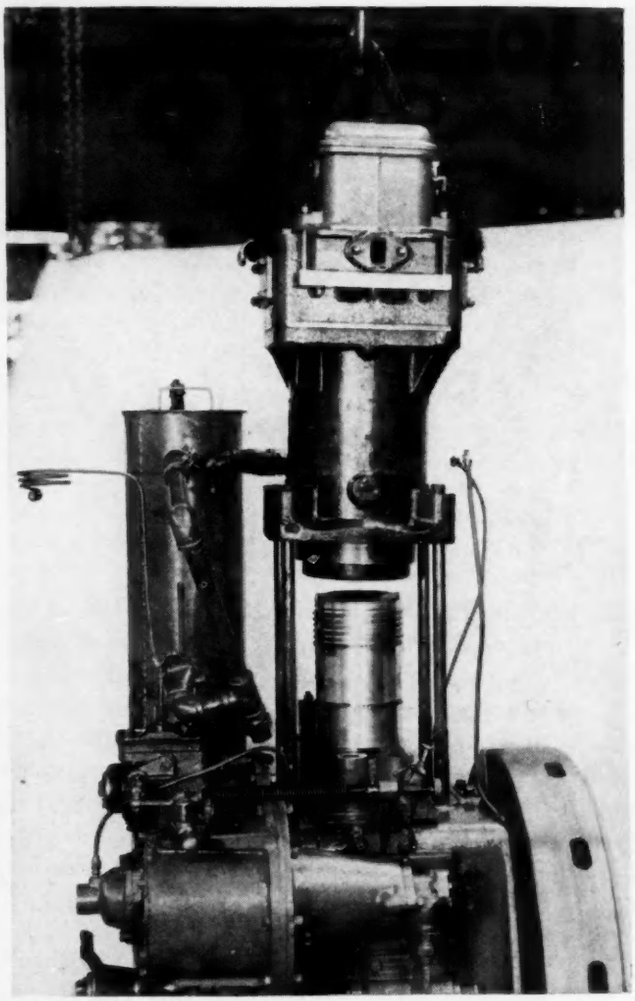


Fig. 2 – Single-Cylinder Diesel Test Engine Dismantled for Inspection of Piston and Rings

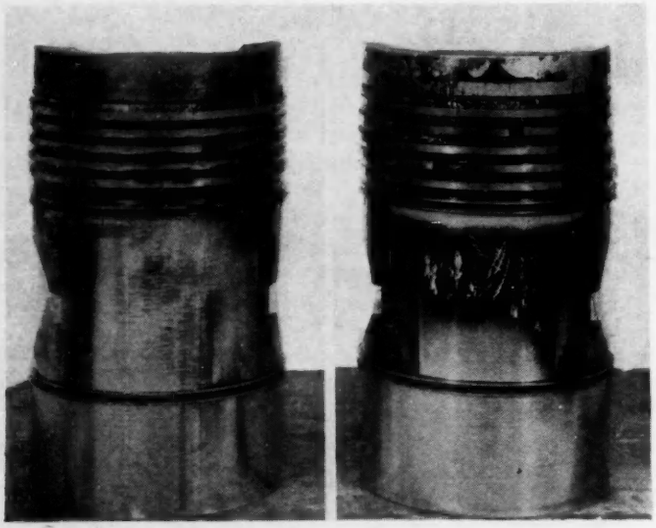


Fig. 1 – Influence of Elevated Ring-Belt Temperatures on Deposits from Lubricating Oils of High Paraffinicity – (*left*) Elevated Ring-Belt Temperature; (*right*) Normal Ring-Belt Temperature

envelope of the combustion gases should permit of a temperature gradient that is conducive to the proper spreading and distribution of lubricant on the cylinder walls and, furthermore, that allows for the desirable continuous movement and flow of the cylinder-wall lubricant for the purpose of promoting good flushing of the products of decomposition. As Boerlage puts it, "Every engine should be self-cleaning."

In the last analysis, gummy deposits, whether from the chemical byproducts of the combustion of fuel or from the decomposition of lubricating oil, function as binders to congeal carbon and dust in the ring-grooves, and ultimately they will produce ring-sticking. It is apparent readily from this explanation that the mechanical structure of the piston and rings will have a bearing upon the influence of these gummy deposits in sticking the rings. Improved formation of ring structures and ring-groove design can effect a method of relief in forestalling the time of ring-sticking. It is within the realm of possibility that piston temperatures can be controlled and regulated so as to render these gummy materials in the ring-grooves less viscous or more completely volatilized.

An example of this control can be found in Fig. 1, which illustrates a piston (*left*) operating at from 80 to 100 deg. fahr. higher ring-belt temperature than found in normal operation (*right*). The higher ring-belt temperature permitted of ring

freedom, whereas the lower temperature range developed serious ring-sticking. Even though this case may be cited as fact, it still remains that, as long as gummy deposits find lodgment on ring-groove lands, the danger of ring-sticking is ever-present and only awaits the thermal environment to produce deposits in such quantities as to destroy piston seal and cause failure of the reciprocating unit. Furthermore, it is a proved fact that minimum cylinder wear on any reciprocating structure is attainable only when the piston-rings float with complete freedom and are responsive instantaneously to the cylinder contour.

The manner in which raw machined surfaces are transformed into bearing surfaces capable of maintaining an oil film is of considerable importance in relation to the ultimate life of the liner and rings. It has been found that breaking-in methods, even over short periods of time, endow reciprocating surfaces with a wear resistance that continues its influence after thousands of hours. Two engines of exactly the same bore, stroke, and speed were broken in on a 14-hr. schedule and then operated for 1000 hr. on the same type of lubricating oil. In the case of one engine, however, an additive was introduced into the lubricating oil during the break-in period only. The influence of this additive for the short break-in period was sufficient to reduce the wear at the end of 1000 hr. to 28 per cent of that of the other engine. By improved break-in methods, the engines are endowed with longer life and better performance.

After having accommodated the Diesel design to improvements dictated by the study of the combustion process and having provided the most suitable and yet practical mechanical structure, it became apparent that lubricants of themselves played a most important role in ring-sticking. Studies in the field on operating units indicated such wide variation in the number of hours required to produce ring-sticking that no

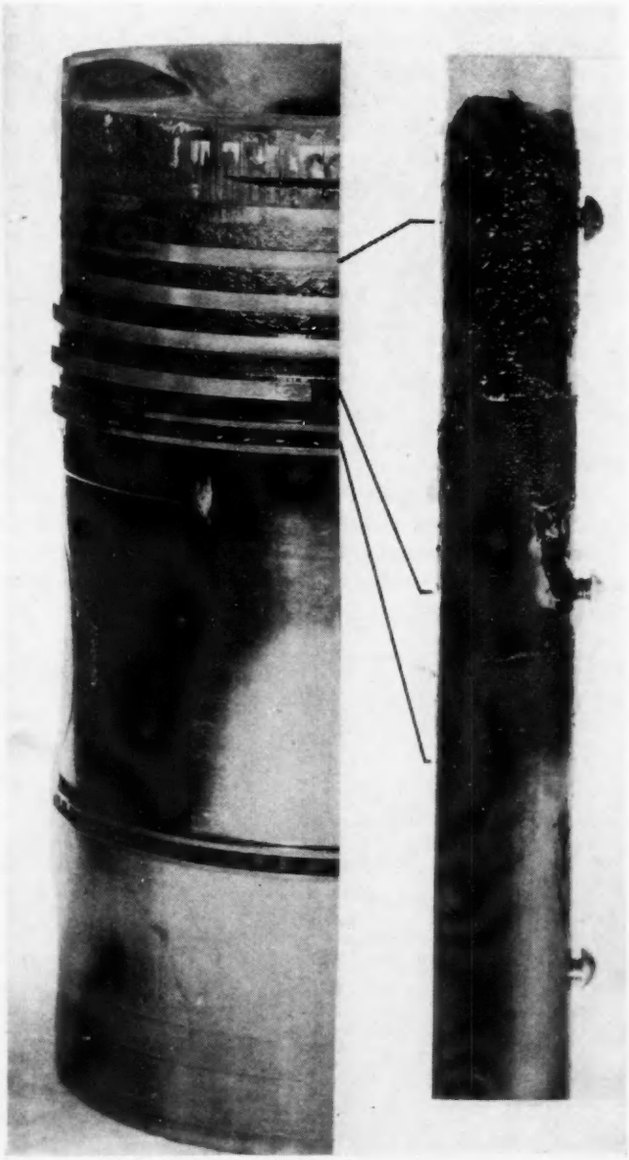


Fig. 3 – Depositions on Piston and Test Bar for Special Highly Treated Type of Lubricating Oil

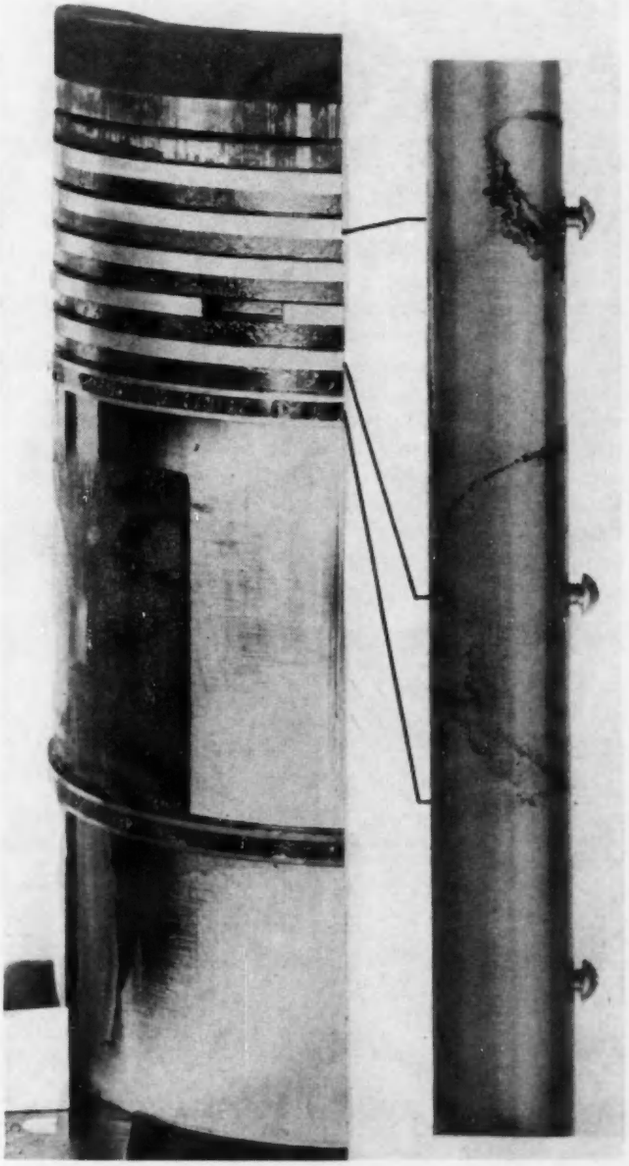


Fig. 4 – Depositions on Piston and Test Bar for Special Acid-Treated Type of Lubricating Oil

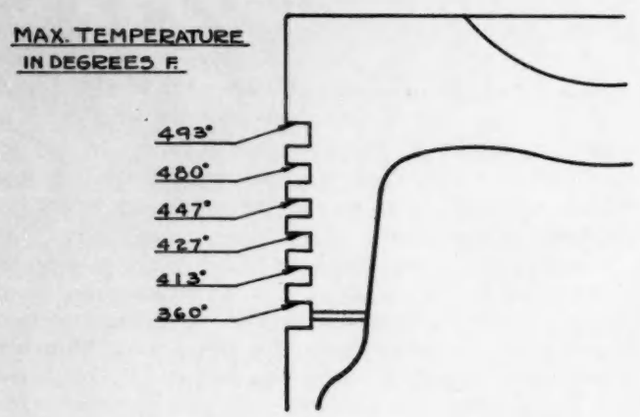


Fig. 5 – Diagram of Temperatures in Ring-Belt Section of Diesel Piston at Normal Full Load

definite conclusions could be drawn except that continuous high load yielded shorter hours of operation than intermittent variable-load operation. Furthermore, some lubricants permitted of higher average hours of operation under high-load service than could be obtained with others.

To further the study of ring-sticking in the laboratory, a single-cylinder Diesel test engine was designed, using production parts, in such a way as to permit of rapid inspection of the piston and rings. Fig. 2 illustrates this unit with the cylinder and cylinder-head withdrawn above the piston without dismantling the reciprocating unit and without in any way disturbing the deposits or petroleum products that have collected in operation on the piston-rings and liner.

Units of this design were operated at the continuous loads that produced ring-sticking in the field in the shortest period of time. It was known that ring-sticking was developed by the accumulation of gummy deposits in the ring-grooves with time. Time, therefore, was a most important factor. This conclusion has been amplified further, since a number of experimental pieces of apparatus have been built to develop ring-sticking by accelerated methods. Accelerated tests have not given satisfaction in checking correlation with field results. The long-distance operation of single-cylinder engines has related ring-sticking in terms of hours of operation before shutdown which, after all, is the most practical measure of this particular problem.

For the sake of convenience the term "ring-sticking hours" will be used in this paper as the measure of time required to cause rings to stick to the extent that dangerous blowby is developed. In the laboratory tests, operation of the ring-sticking test engines was continued until the blowby of the gases from the crankcase reached values of 85 cu. ft. per hr. Normal operation is characterized by a blowby of 12 to 20 cu. ft. per hr. in the single-cylinder unit. Tests were stopped at the high-blowby point, as this amount indicated serious ring-sticking. It was desired further that all evidence of deposition be saved. Continued operation might result in the destruction of valuable information as to surface conditions and petroleum deposits.

A study of the operation of a number of lubricants, generally considered suitable for Diesel engines, in the single-cylinder test engine revealed the startling information that ring-sticking would occur as early as 100 hr. of operation on one type of lubricant, or as late as 3000 hr. on another type. In all these tests the engine conditions relative to the control of temperature of the combustion-chamber envelope were maintained constant. This significant finding naturally pointed conclusively to the point that lubricating oils of themselves influenced ring-sticking. Further analysis of these tests indicated that the ring-sticking hours could be tabulated in three major divisions when the lubricants were related to methods of treatment and refining.

In the present state of flux in the manufacture of lubricating oils it is perhaps unwise to draw hard and fast lines concerning these findings. In the state of the art two years ago it was apparent that ring-sticking hours exhibited a relationship in inverse ratio to the degree of treatment inflicted upon the oil stock. Drastic treatment developed low ring-sticking hours. Minimum chemical change during finishing was conducive to

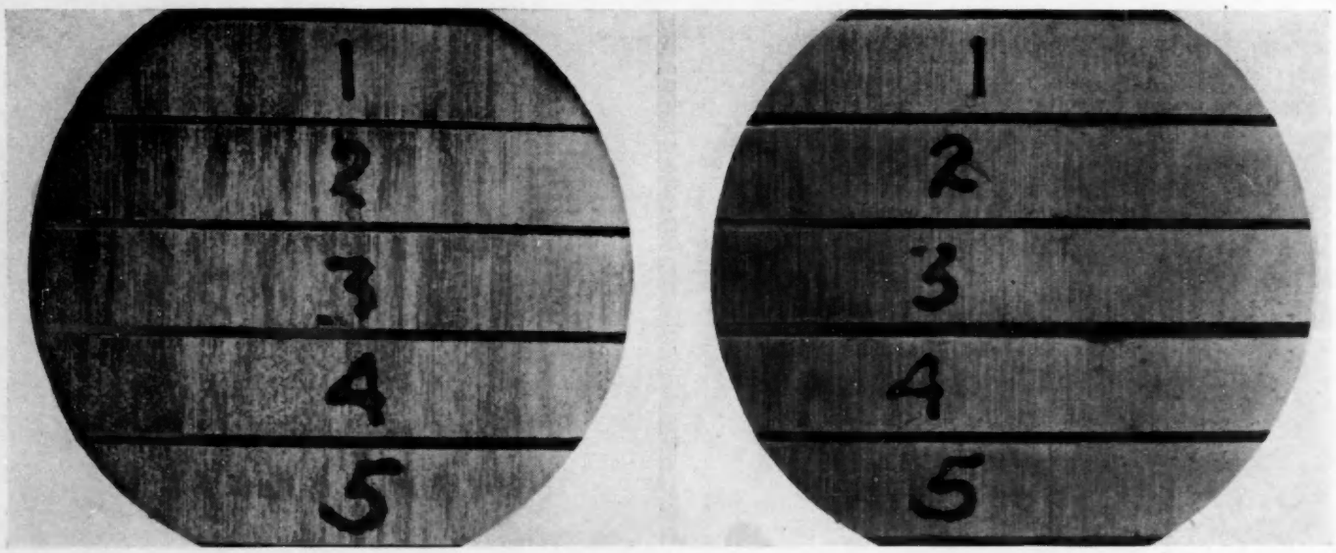


Fig. 6 – Influence of Additive for Improving Film Strength – (*left*) Straight Mineral Oil; (*right*) Straight Mineral Oil Plus Film-Strength Compound

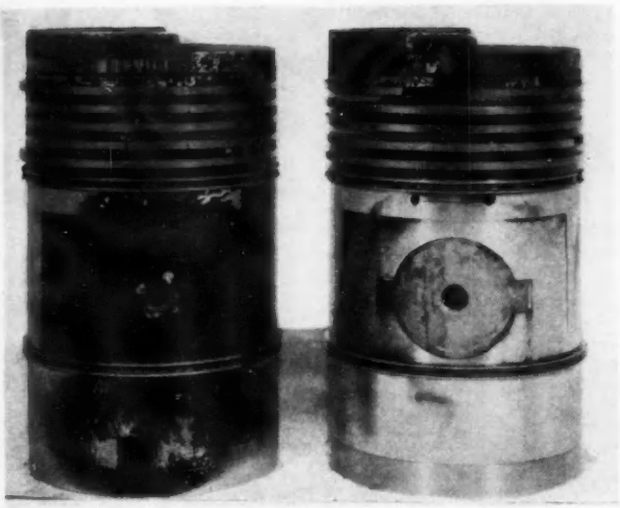


Fig. 7 – Influence of Compounds To Improve Ring-Sticking Hours – (*left*) Straight Mineral Oil; (*right*) Compounded Oil

high hours of operation before ring-sticking developed failure.

It was also evident that, under the temperature gradient prevailing in the ring-belt section, the source of the base stock was significant in its influence on ring-sticking hours. Naphthenic-base oils showed the highest ring-sticking hours in each division. Results obtained more recently at elevated ring-belt temperatures show partiality toward the more stable lubricants of high paraffinicity. Such temperatures, however, are not maintainable under broad-range service operation.

Fig. 1 expresses the influence of higher ring-belt temperature on ring-sticking on a special highly solvent-treated paraffin-base lubricating oil. The piston on the right was obtained when operating a standard piston under normal ring-sticking-test conditions. The illustration on the left is of a specially designed piston in which the ring-belt section was controlled thermally so as to develop ring-groove temperatures from 80 to 100 deg. fahr. higher than normal. The latter illustration speaks for itself and indicates the freedom of rings at the elevated temperature, together with far less deposition over the entire piston.

These investigations and results led to the study of oxidation and coking tests. Most outstanding of all these tests in their bearing upon piston-ring-belt deposits, were coking tests made upon metals that were subjected to a controlled heat gradient. These tests showed that mineral oils exhibited considerable variation in the type and quantity of deposits formed, ranging in consistency from the hard, tenacious lacquers to the soft, spongy graphites. The base stock and treatment in refining influenced the extent of the temperature limits within which these deposits were formed. Some oils allowed products of decomposition within narrow temperature ranges, whereas others spread over wide regions of temperature.

These results were not directly parallel to piston deposits taken from engine tests, but exhibited a proportional relationship which explained much of the mystery of ring-sticking hours. For instance, it was known that, under the conditions of the ring-sticking tests, paraffin-base oils would have a tendency to stick the top ring first, whereas naphthenic-base oils would cause the seizure of lower rings on the ring-belt section.

The wide differences in piston depositions can be observed by referring to Figs. 3 and 4. Fig. 3 illustrates a piston and a test bar taken from a special highly solvent-treated paraffin-base oil. The deposits accumulated on this piston, sufficient to cause ring-sticking of the three top rings, were developed in a little over 100 hr. The test bar indicates that dangerous products were formed between a range of 350 and 510 deg. fahr. In correlating the bar test with the actual piston deposits, the chart shown in Fig. 5 served the purpose of identifying these products in specific temperature ranges. Figs. 3 and 4 are of importance when considered comparatively.

In Fig. 4 the deposits accumulated on the piston surface required a little under 2000 hr. when using a special acid-treated naphthenic-base oil. At this point none of the rings were stuck. The test bar, although not colored and therefore not illustrating the yellow and orange gums, illustrates the narrow temperature range within which deposits are formed. The quantity of deposit is extremely small for the bar test, and one would naturally expect high operating hours from this base stock of oil. These bar tests are of considerable value in checking base stocks, but they are not as significant when checking compounded oils.

With information of the character just described, it would be conceivable that a Diesel piston could be designed that would have a thermostatically controlled ring-belt temperature gradient sufficiently outside of the deposition range of the particular mineral oil selected so that binder-type products would not be formed and so that operation could be extended without serious danger from ring-sticking. For instance, in the case of Fig. 3, ring-belt temperatures of 520 deg. fahr. and above should insure freedom from deposits that cause ring-sticking. It also would be possible, although not practical, to develop pistons that would operate at high loads at ring-belt temperatures below 350 deg. fahr., in which case there would be little likelihood of danger of producing ring-sticking.

A different type of mineral oil would call for an entirely different piston-ring-belt temperature gradient for maximum ring-sticking hours. Such designs of pistons, however, are not practical or economical when considering wide-range service conditions on engines of wide geographic distribution. It might be opportune here to explore the reverse side of the picture.

In field operation in certain geographic locations under specific service conditions, records have been established in which Diesel-tractor operation has extended well over 10,000 hr. on straight mineral oils without serious ring-sticking. These same straight mineral oils, however, have been known to yield failures due to ring-sticking at very low hours of operation in other territories and under entirely different service conditions.

TYPE OF OIL	RATE OF WEAR PER 1000 HRS. AT MAX. POINT	
	LONG.	TRANS.
OIL 'X' BASE	.0072	.0079
OIL 'X' + COMP	.0029	.0026
OIL 'Y' BASE	.0036	.0036
OIL 'Y'+COMP.	.0006	.0008

Fig. 8 – Comparative Liner Wear of Straight Mineral Oils Vs. Compounded Oils

This situation, when viewed in the light of the influence of piston-ring-belt temperature on the deposition of gummy and carbonaceous matter in the piston-ring grooves, leads one to the conclusion that the development of outstanding lubricants preeminently suited to cope with Diesel-engine conditions might offer greater alleviation to the ring-sticking problem than super-controlled and sensitive piston-ring-belt structures. It is also conceivable that suitable base stocks of desirable treatment and finish when alloyed with ingredients endowed with capacity to deal with special problems, should combine to offer a sum total in operating results of great superiority.

Alloyed steels made possible the modern automobile. Alloyed lubricating oils point the way to improved engine performance and greatly reduced maintenance. This statement is no idle gossip. Test results and operating data definitely substantiate increased life of rings and cylinders and longer uninterrupted service between shutdowns for repairs of the reciprocating parts. Alloys and compounds improved base stocks to yield outstanding Diesel-engine lubricants by accentuating such desirable properties as:

(1) Increased film strength.

(2) Increased ring-sticking hours, almost to the practical elimination of this evil.

(3) Increased life of cylinders and rings.

(4) Reduction in sludging tendency.

(5) Improved cylinder sealing and consequent reduction of blowby of gases into the crankcase.

(6) Improved flushing and cleansing action on piston thrust surfaces, in ring-grooves, and upper cylinder wearing surfaces.

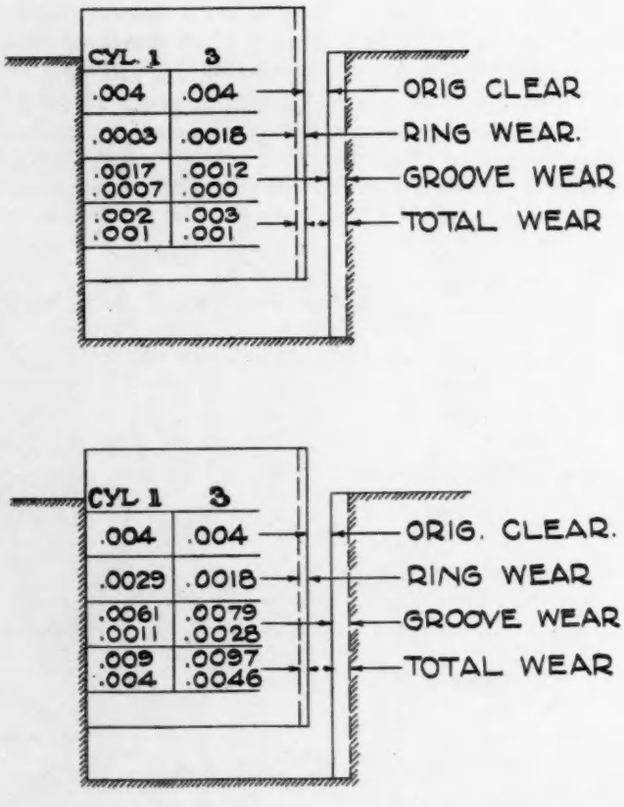


Fig 9 – Ring and Ring-Groove Wear – Showing Advantage of Compounded Diesel Lubricating Oil

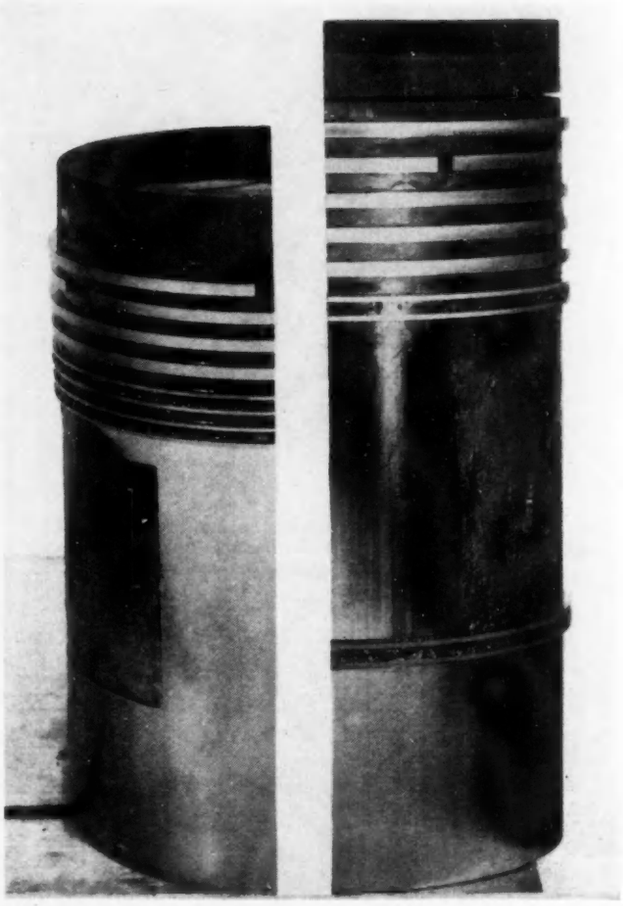


Fig. 10 – Influence of Compounds on Sludge Depression – (*left*) Compounded Oil; (*right*) Straight Mineral Oil

(7) Stabilization of efficient oil consumption without fear of forming deleterious products of decomposition.

To illustrate these important contributions of alloys and compounds in lubricating oils, consider first the need for increased film strength. Fig. 6 illustrates the conditioning received by the piston-rings in comparable break-in tests. The rings on the left used a straight mineral oil somewhat lacking, because of treatment, in sufficient film strength to permit of 100 per cent successful break-in without scratching. During the early conditioning period of break-in, it is incumbent that the lubricant separate the rings from the cylinder walls under the severe load condition attained when the full cylinder pressure is directly behind the top ring. The lubricant used in the test of the left-hand piston was not capable of maintaining a film during the transition period of the break-in.

The right-hand rings of Fig. 6 are from the break-in run of a lubricant consisting of the same base stock as was used in the left-hand piston, but to which was added an ingredient improving film strength. No surface abrasions on the rings are evident in the latter case shown on the right.

Under the severe operating conditions of tractors in all sorts of weather and for all manner of service, it is essential that added film strength be available for emergencies. The starting of engines after long periods of shutdown in weather conditions that are most severe for initiating lubrication in the hands of operators prone to abuse equipment, presents severe penalties on the immediate establishment of suitable oil films.

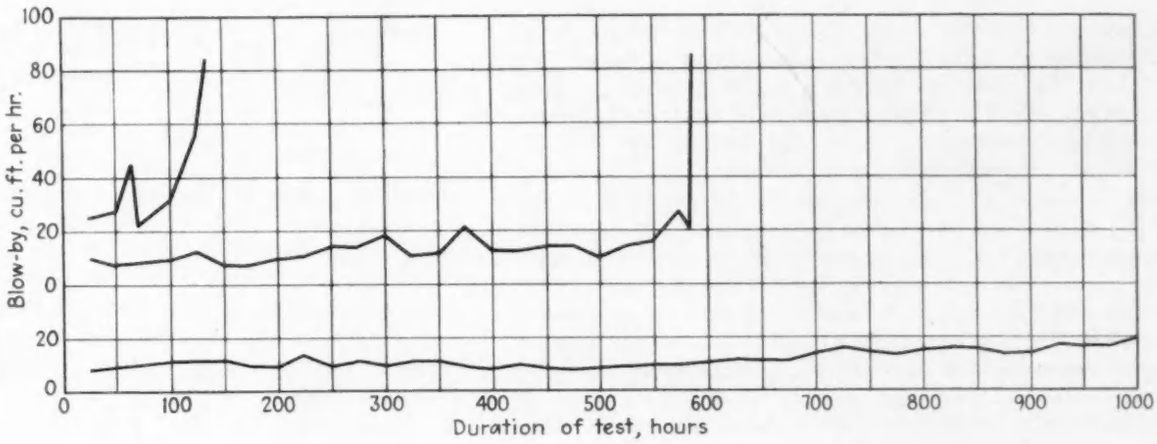


Fig. 11 – Blow-by Curves Taken from Three Distinct Types of Lubricating Oils

The most striking example of improvement by increasing ring-sticking hours of straight mineral oils by the addition of compounds is illustrated in Fig. 7. The left-hand piston is taken from a run of less than 1000 hr. when using an SO_2 treated Western-base straight mineral oil. To this base stock was added a very small percentage of material to develop resistance to ring-sticking. The piston on the right is taken from a run of comparable hours using the compounded oil, and illustrates generally the cleanliness of the piston and its elements. The rings are entirely free, no deposits are attached to the piston thrust surfaces, the oil rings are clean and open, and the piston-pin relief still shows the machined surface. In the case of the piston operating on the straight mineral oil (shown at the left), the top ring is stuck 50 per cent, the second, third, and fourth rings are stuck 100 per cent, the oil ring-grooves are badly sludged, and the piston-pin relief and thrust surfaces are coated with heavy, gummy deposits. The right-hand photograph illustrates a typical case where ring-sticking hours are extended to a point where overhaul will be occasioned by failures other than ring-sticking.

One of the most interesting improvements incident to the use of compounded oils is cited in the reduction of cylinder-liner wear. With good lubricants tested in the single-cylinder test engine, it is customary to expect a maximum rate of cylinder wear at the top of the ring travel of 0.0025 to 0.004 in. per 1000 hr. after 1000 hr. of operation. Some mineral oils have reached rates of 0.009 to 0.011 in. per 1000 hr., whereas some of the new alloyed lubricants have recorded as low wear as 0.0003 to 0.0005 in. per 1000 hr. at the 1000-hr. period.

Take two specific examples as tabulated in Fig. 8. Here the wear of the base-stock oil is compared at 1000 hr. of operation with the wear occasioned by the base stock plus an anti-ring-sticking compound. Oil *X* is a Western-base oil, and oil *Y* also is a naphthenic-base lubricant. In the case of oil *X* the liner wear is cut to less than half, whereas, in the case of oil *Y*, the liner wear is cut approximately 1/5. Free-floating piston-rings are a most necessary prerequisite to low liner wear. Surely these facts are significant and provide desirable savings readily appreciated in commercial operation.

A further point of interest is expressed in the out-of-round condition often found when using some straight mineral oils. The table in Fig. 7 is not convincing in this respect, but high-gumming oils cause piston-rings to become tacky and bind in localized regions. This condition produces out-of-round wear which may reach values as high as 0.005 to 0.009 in. out-of-round. This situation presents a serious problem in piston sealing.

The improvement in ring-groove wear by the use of alloyed or compounded oils is expressed in the table of Fig. 9. Here it is revealed that low-wear characteristics are inherent in all the wearing surfaces of the reciprocating rubbing parts in the engine.

Of considerable interest is the reduction in sludging tendencies, particularly of Western-base oils. As a matter of fact, the use of dopes or additives performs most effectively in depressing sludge formation. Take Fig. 10, for instance, wherein two pistons are shown, each of which employed the same base stock of Western oil as a lubricant. The sludge

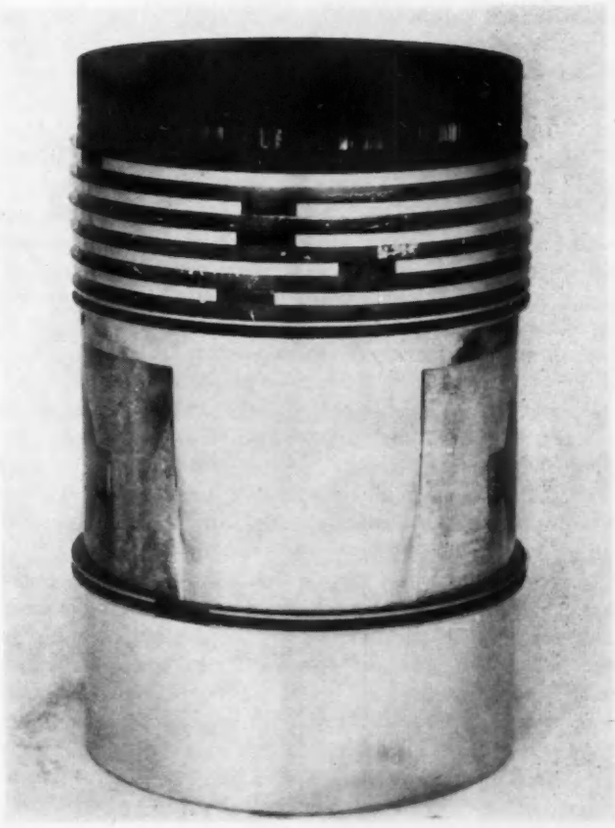


Fig. 12 – Influence of Flushing and Cleansing Action of Alloyed Lubricating Oils on Piston Appearance

deposits at 1000 hr. of the undoped oil are evident in the right-hand photograph. Here accumulations in the oil ring-groove, the piston-pin reliefs, and the thrust surfaces are unmistakable. The left-hand photograph was taken after 1500 hr. of operation with the doped oil of the same base stock and illustrates the remarkable absence of the deposits in the regions mentioned.

Perfectly free-floating piston-rings not only reduce wear but provide the much-desired piston-sealing properties so essential to good Diesel operation. A heavy type of deposit in the piston-ring belt and on the piston skirt may be conducive to high sealing value against blowby gases but of short duration, whereas free-floating rings will permit of a certain amount of blowby through more open areas in the ring-grooves, but the consistency of blowby never can be preserved with the high-gumming oils, whereas the non-gumming oils will maintain a very uniform blowby rate.

This point is illustrated best by Fig. 11, wherein three oils are compared as to blowby in ring-sticking endurance runs. The top curve is from a lubricant of the character shown in Fig. 3, yielding high piston deposits. The middle blowby curve is illustrative of a test conducted on the piston on the left-hand side of Fig. 7. The lowest curve is for a piston as shown on the left-hand side of Fig. 10. These curves and photographs tell their own related stories. The sticking and unsticking of rings is indicated by peaks and valleys in the blowby curve. Free-floating piston-rings preserve a uniform rate of blowby and iron out the irregularities in the blowby curve. Generally speaking, the character of the blowby curve is of greater significance than the actual blowby rate.

Compounded or alloyed lubricating oils, endowed with cleansing properties, permit of the flushing and washing away of materials that have a tendency to deposit in ring-grooves or on piston clearance surfaces. The net result is a clean piston, even after thousands of hours of operation, as illustrated in Fig. 12.

Where rings are lubricated effectively and are absolutely free-floating, there is a tendency toward the stabilization of efficient oil consumption. By an efficient oil consumption is meant sufficient lubricant to resist wear in such regions as the top of the piston-ring travel. With straight mineral oils, which have a tendency toward producing dangerous gummy deposits, lubricating-oil consumption is usually curtailed to a minimum. The fear of the formation of deleterious products of decomposition places restrictions on the amount of lubricant permitted to reach the upper portions of the cylinder surface. When using lubricants that are free from dangerous depositions, there is no necessity for the restriction of an ample supply of lubricant to all cylinder surfaces. The use of alloyed or compounded lubricants greatly facilitates copious lubrication.

In conclusion, it may be said that alloyed and compounded lubricants have produced outstanding results, both in the laboratory and in the field. They are capable of going into practically all regions and territories and types of service, and maintain consistency of performance without dependence upon sensitively controlled temperature gradients. The present-developed special Diesel-engine lubricants may be intermediate in their development stage, but they point the way toward improvements in lubricants to qualify for exacting service conditions. Their value in promoting superior field performance and lowering maintenance costs already has justified their existence.

A Spark-Plug Adapted for Measuring Cylinder-Head Temperatures

(*Continued from page* 152)

pensated by the fact that the data obtained are more reliable. Plugs equipped with this device have been in use for several months at the Delft Laboratory. So far, no failure of any part of the thermocouple or its insulation in the plug body has occurred. Neither has any change in the characteristics of the spark-plug, or in its reliability, been observed.

Tests on engines. – In Figs. 4 and 5, comparative measurements with the three methods are shown.

The three devices were incorporated in one spark-plug, and readings were taken every 15 sec. simultaneously on three galvanometers.

Determinations on a *water-cooled,* variable-compression engine (Ricardo E 35) are shown in Fig. 4.

It will be noticed that the temperature of the couple in the face of the plug is but little higher than that recorded by the washer couple, which relation may be ascribed to the equalizing effect of the water-cooling. Further, it is seen that the couple in "the face of the plug" shows changes in conditions (detonation intensity, compression ratio) more clearly than does the washer couple.

Determinations on an *air-cooled* engine (J.A.P. motorcycle engine) are shown in Fig. 5.

It is shown that in this engine the temperature of the couple in the face of the plug is approximately 150 deg. cent. higher than that recorded by the washer couple, which relation is explained by the more effective cooling of the latter location.

When the wind speed is increased, for instance from 16.5 to 25 m. per sec., the temperature of the washer couple falls quicker than the temperature of the couple in the face of the plug while the temperature of the central electrode remains constant. Detonation caused only a slight increase of the washer temperature (up to the temperature previously attained at the low wind speed; see point *B* in Fig. 5), whereas both the plug couple and the central-electrode couple indicated a marked increase in temperature. From this observation it may be concluded that, when measuring the temperature of the washer couple, an increase in wind speed can obliterate the much more obnoxious heat rise of the cylinder walls caused by detonation, the latter rise being indicated quite clearly by the thermocouples in the plug and in the central electrode.

Conclusion. – Although the temperature measurement in the face of the plug seems rather delicate for use in aviation routine, it is now generally applied in our laboratory work on account of the following advantages over the spark-plug washer couple and over the central-electrode couple:

(1) The temperature registered by this device is practically equal to that of the combustion-chamber wall and varies accordingly without appreciable time lag.

(2) The insertion of the thermocouple is possible on practically every detachable automotive or aviation plug and is not much more encumbrous than the original plug.

(3) The high tension and the temperature-measuring circuit are separated completely.

(4) The most appropriate materials can be chosen.